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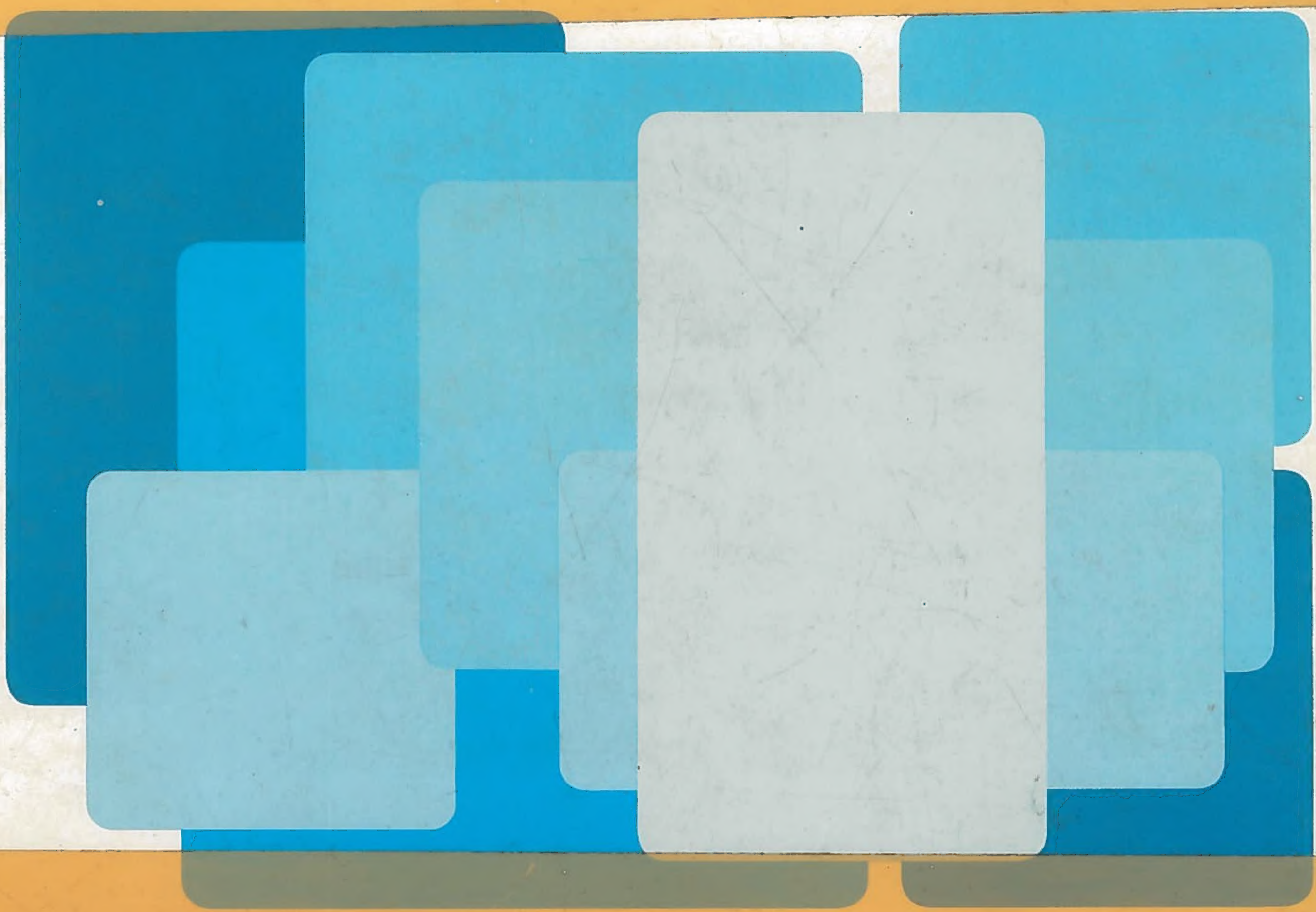
The Hydraulic Trainer



RE 00 301

The
Hydraulic Trainer

Instruction and Information on Oil Hydraulics

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The "Hydraulic Trainer" gives a brief insight into hydraulic technology.

It starts by explaining the physical principles involved, and gives examples of practical application to show the variety of uses possible.

In the chapters following this introduction, the functions and purposes of the individual hydraulic elements are explained, such as:
pumps, motors, cylinders, check valves, directional control valves, pressure control valves, flow control valves, proportional valves, servo valves, accumulators and accessories.

Methods of connection and hydraulic power units are also discussed.

The book concludes with circuit diagrams and calculation formulae.

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Summary of Contents

G. L. Rexroth GmbH

G.L. Rexroth GmbH, based in Lohr am Main, West Germany, is the largest manufacturer of hydraulic equipment in Europe. It has fifteen subsidiary companies, both in Europe and overseas, and is thus represented on a worldwide scale.

The hydraulic programme includes: directional control valves, flow control valves, pressure control valves, check valves, 2/2 directional cartridge valves, vane pumps, radial piston pumps, axial piston motors, gear pumps, hydrostatic transmission, water hydraulics and servo hydraulics.

The products find application in the following industry sectors:

machine tools, plastics machinery, heavy machinery, steelworks, presses, ship construction, offshore and onshore industries, agricultural machinery, mobile equipment, civil engineering, mechanical handling equipment.

Complete systems are designed by a team of qualified engineers. Rexroth developed their first hydraulic valves in 1953.

However, the inspiration for the 'hydraulics business' stems from the foundry. This was founded in 1795 and is today one of the most important in the field of special iron.

There is a wide range of manufacture:

Rexroth HK Special Cast-Iron

Rexroth Hydraulic Cast-Iron

Rexroth Chill-Cast Spheroidal Graphite Cast-Iron GGG 40, 50 and 60

Rexroth Spheroidal Graphite Cast-Iron, cast in sand, GGG 40, 50, 60 and 70 (hydraulic quality)

Rexroth Continuous Casting

Rexroth Special Alloys.

Since January 1976, Rexroth, once a family business, is a wholly owned subsidiary of Mannesmann AG, Duesseldorf. Approximately 3500 people are currently employed in the main works in Lohr/Main.

The Rexroth Group has over 10000 employees and is represented worldwide by its associate companies in the United Kingdom, USA, France, Italy, Switzerland, Spain, Austria, Belgium, Brazil and India.

Rexroth is a dynamic and growing organisation, with a qualified team, high quality products and a sound economic basis.

General

"Hydraulics Made Easy"

With this book, we hope to give you an insight into the world of hydraulics.

Experience has shown that hydraulics is now indispensable as a modern method of transferring energy.

Hydraulic drives and controls have become more and more important due to automation and mechanisation.

Today, a very large amount of modern and powerful machinery is controlled either partly or completely by hydraulics.

The use of hydraulic systems for control and regulation has made possible important fields for automation.

However, in spite of the wide range of application, the knowledge of this specialised field has not yet been circulated to a high enough degree. As a result of this, the application of hydraulic systems has been restricted.

Whoever decides to use hydraulics for power transmission and movement, however, is not treading on new ground.

We have a great deal of experience.

However, the knowledge of the elements of physics and an understanding for the processes of hydraulic equipment and systems has led to this experience, so that this basic knowledge is a necessary pre-requisite.

We are sure that many points will appear clearer and easier to understand, after careful study of this book.

Basic Principles

Before we look at hydraulics in detail, we must firstly define what is meant by this term.

The word "hydraulics" is derived from the Greek word "hydor" and means "water". This comprised all things in affiliation with "water".

Today, we understand under the term "hydraulics" the transmission and control of forces and movement by means of fluid.

Fluid is therefore used to transmit energy. Mineral oil us used generally, although a synthetic fluid, water, or oil-water emulsion can be used.

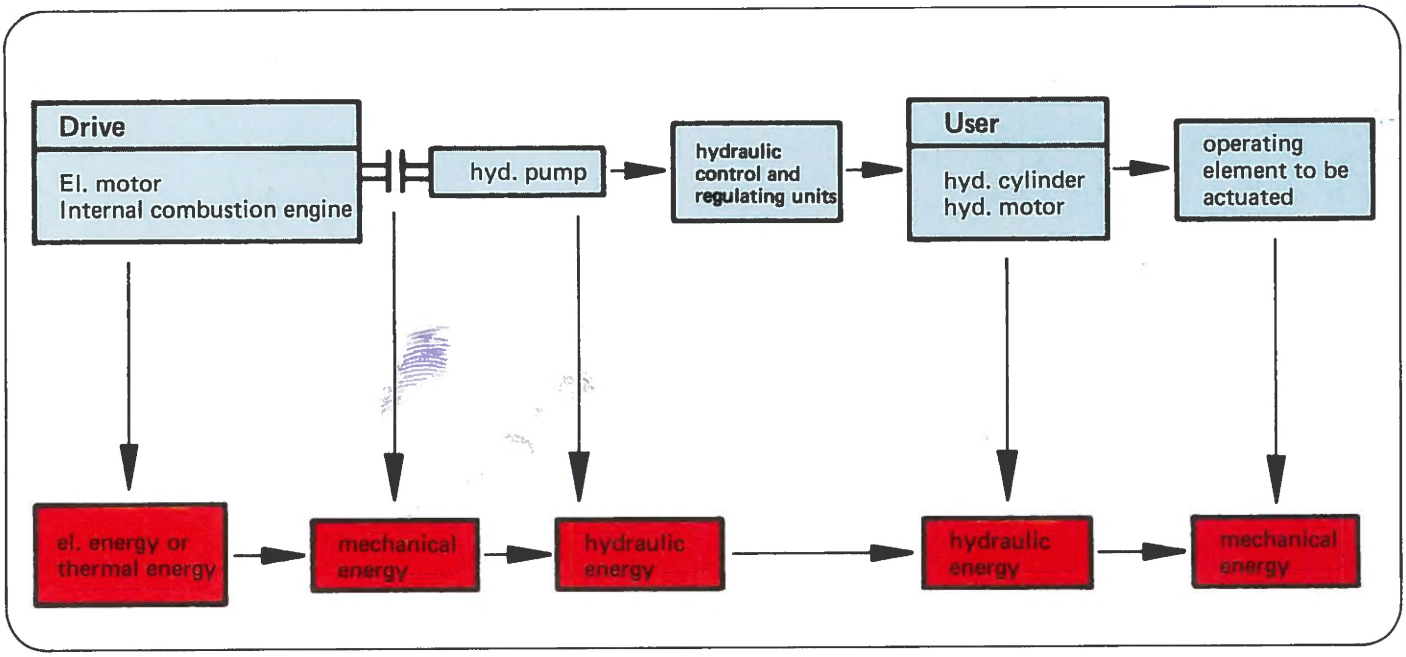
The field of hydromechanics (fluid mechanics) is broken down as follows:

Hydrostatics	mechanics of still fluid (theory of equilibrium conditions in fluids)
Hydrodynamics	mechanics of moving fluid (flow theory)

An example of pure hydrostatics is the transfer of force in hydraulics.

An example of pure hydrodynamics is the conversion of flow energy in turbines in hydro-electric power plants.

Conversion of energy in a hydraulic unit.



Basic Principles

There are, of course, other methods of energy transmission apart from hydraulics, e.g.

mechanical (gear, shaft, crank mechanism, etc.)

electrical (rotating field motor, linear motor or moving field motor, torque motor, etc.)

electronic (amplifier, electronic conversion elements)

pneumatic (similar to hydraulics, air is used as transfer element).

Each of the above has its own fields of application. However, in some cases, it is possible to select from the various possibilities.

There are many points in favour of a hydraulic control and hydraulic drive.

Special characteristics of hydraulics:

- High forces (torques) with compact size, i.e. high power density.
- Automatic force adaption.
- Movement from standstill possible under full load.
- Stepless change (control or regulation) of speed, torque, stroke force, etc. can be achieved simply.
- Simple overload protection.
- Suitable for controlling fast movement process and for extremely slow precision movements.
- Relatively simple accumulation of energy by means of gas.
- Combined with decentralised transforming of the hydraulic energy back into mechanical energy, simplified central drive systems are possible, giving a high degree of economy.

Basic Principles

Weight, Pressure, Force

Definitions and calculations for the international unit system (SI units).

A **mass** (to be understood as a material quantity) of 1 kg creates a weight force of 1 kp on the ground.

According to Newton's law:

$$F = m \cdot a$$

Force = mass x acceleration

$$Kg = \frac{m}{s^2}$$

According to the old system, with acceleration due to gravity g for general acceleration a :

$$F = m \cdot g$$

$$1\ Kp = 1\ Kg \cdot 9.81 \frac{m}{s^2} = 9.81 \frac{kg \cdot m}{s^2}$$

According to the SI unit system, force F is stated in Newton units (N)

$$1\ N = 1\ Kg \cdot 1 \frac{m}{s^2} = 1 \frac{kg \cdot m}{s^2}$$

thus $\boxed{1\ kp = 9.81\ N}$

In practice, the following is generally adequate:

$$1\ kp \approx 10\ N = 1\ daN$$

Pressure, one of the most important measurements in hydraulics, is defined as force per area.

$$\boxed{p = \frac{F}{A}}$$

p = pressure in bar
F = force in N
A = area in cm^2

Previously, pressure was stated as $\frac{kp}{cm^2}$

$$1 \frac{kp}{cm^2} = 1\ at\ (1\ atmosphere)$$

As Newton units are now used for force, the following results:

$$\boxed{1\ bar = 10 \frac{N}{cm^2} = 1 \frac{daN}{cm^2}}$$

$$1\ bar = 1.02 \frac{kp}{cm^2}$$

$$1 \frac{kp}{cm^2} = 0.98\ bar$$

If the fundamental quantities for force (N) and area (m^2) according to SI units are used, the Pascal (Pa) unit is used for pressure.

$$1\ Pa = 1 \frac{N}{m^2}$$

However, since the Pascal unit results in practice in values which are too high, it is preferable to use the Bar unit (bar).

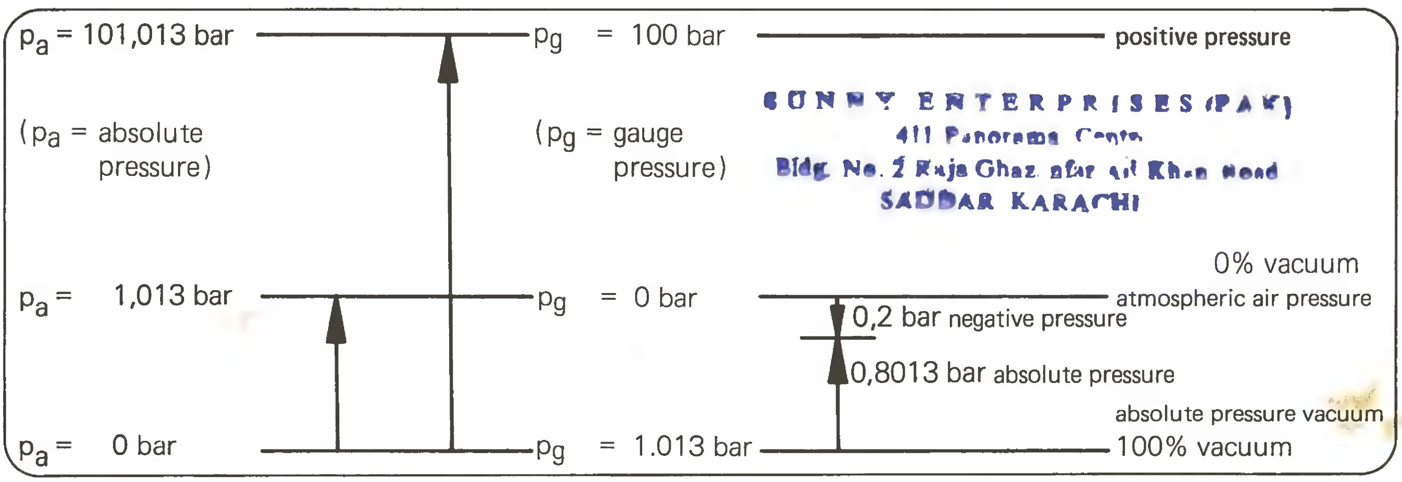
$$\boxed{1\ bar = 100\,000\ pa}$$

The unit psi (pound force per square inch) is still used

1 bar = 14.5 psi

NB: this unit is not included in the SI unit system.

Pressure details in (bar) according to SI units refer to absolute pressure.



Basic Principles

In hydraulics, operating pressure is generally stated as p. This refers to the excess pressure above atmospheric level.

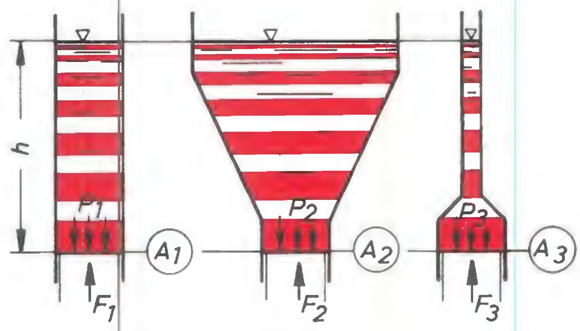
Hydrostatics (mechanics of still fluids)

Hydrostatic pressure (gravity force)

Inside a head of liquid, pressure occurs due to the weight of the fluid mass over a determined area. The pressure is related to the height of the head of liquid (h), the density (ρ) and the accelation due to gravity (g).

Gravity force $p = h \cdot \rho \cdot g$

fig. 1



Taking tanks of different shapes filled with the same fluid, the pressure at one certain level is related only to the height of the head of liquid.

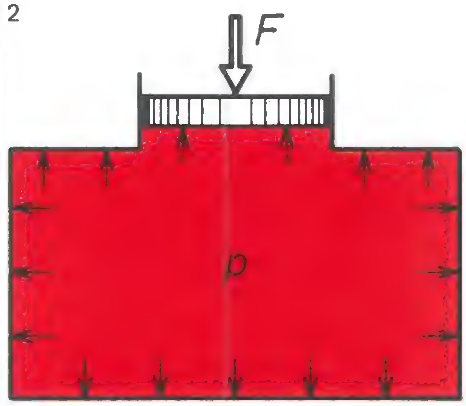
$p_1 = p_2 = p_3$ (fig. 1)

Hydrostatic pressure creates a force on the base of the tank.

If the pressure as shown in fig. 1 acts on the same surface area in each container ($A_1 = A_2 = A_3$), the forces resulting from this are also equal ($F_1 = F_2 = F_3$).

Pressure by External Forces (Pascal's Law)

fig. 2



Where force F acts on an enclosed fluid via surface A (fig. 2), pressure occurs in the fluid.
The pressure is related to the amount of force applied to the surface vertically and the area.

$$\boxed{p = \frac{F}{A}}$$

p in bar
F in N
A in cm²

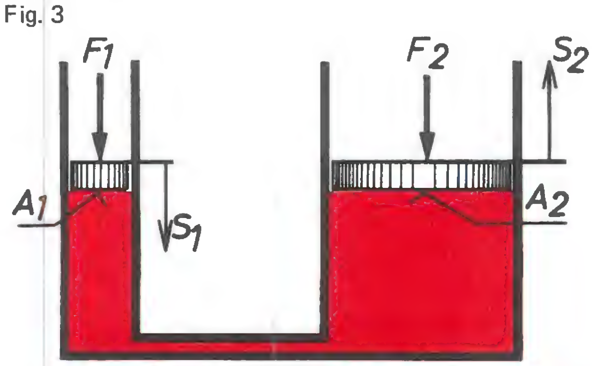
Pressure acts on all sides equally and simultaneously. It is therefore equal at all points. This is valid with omission of the gravity force, which would have to be added according to the fluid level.

Due to the pressure at which hydraulic systems operate, this fraction need not be considered.

e.g. 10 m head of water ≈ 1 bar

Hydraulic Force Transmission

Fig. 3



$$\frac{S_1}{S_2} = \frac{A_2}{A_1} = \frac{F_2}{F_1}$$

Basic Principles

As the pressure distributes equally to all sides, the shape of the tank is not important. In order to operate with pressure created by an external force influence, we use a system as shown in fig. 3.

If we now pressurise surface A_1 with force F_1, we create pressure

$$p = \frac{F_1}{A_1}$$

Pressure p affects all parts of the system, therefore also on surface A_2. The force which can be achieved (tantamount to a load to be lifted).

$$F_2 = p \cdot A_2$$

$$\text{thus} \quad \frac{F_1}{A_1} = \frac{F_2}{A_2}$$

$$\text{or} \quad \boxed{\frac{F_2}{F_1} = \frac{A_2}{A_1}}$$

The forces have the same relationship to each other as the surfaces.

The pressure in such a system always depends on the size of the load and the effective surface.

This means that the pressure rises until it can overcome the resistance, which builds up in opposition to the fluid movement.

If it is possible to achieve the pressure necessary to overcome the load F_2 (via surface A_2) by means of force F_1 and surface A_1, then load F_2 can be raised. (Friction losses need not be taken into consideration here.)

The relationship of the paths S_1 and S_2 of the two pistons is then opposite to that of the surfaces.

$$\boxed{\frac{S_1}{S_2} = \frac{A_2}{A_1}}$$

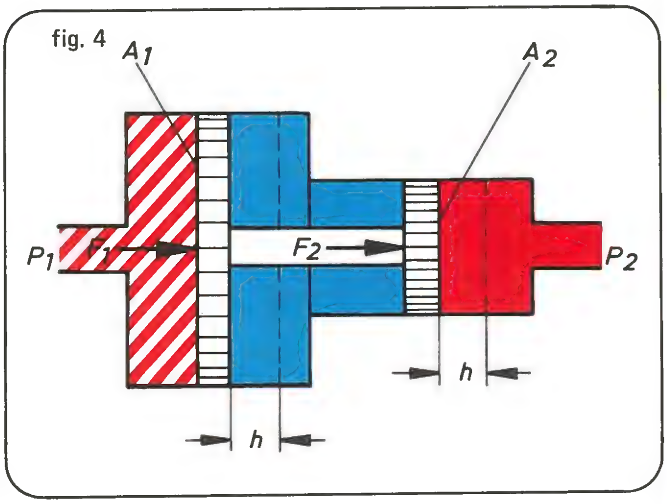
The function of the force piston W_1 is the same as that of the load piston W_2.

$$W_1 = F_1 \times S_1$$

$$W_2 = F_2 \times S_2$$

Principle of Pressure Transmission

fig. 4



Two pistons of different sizes are fixed rigidly to one another by means of a piston rod. If surface A_1 is affected by pressure p_1, force F_1 occurs at the large piston. Force F_1 is transferred to the smaller piston by means of the piston rod. This force now affects surface A_2 and causes pressure p_2 (fig. 4). The following are valid without friction losses:

$$F_1 = F_2 = F$$

$$p_1 \cdot A_1 = p_2 \cdot A_2$$

$$\text{thus} \quad p_1 \cdot A_1 = F_1$$

$$p_2 \cdot A_2 = F_2$$

$$\text{or} \quad \frac{p_1}{p_2} = \frac{A_2}{A_1}$$

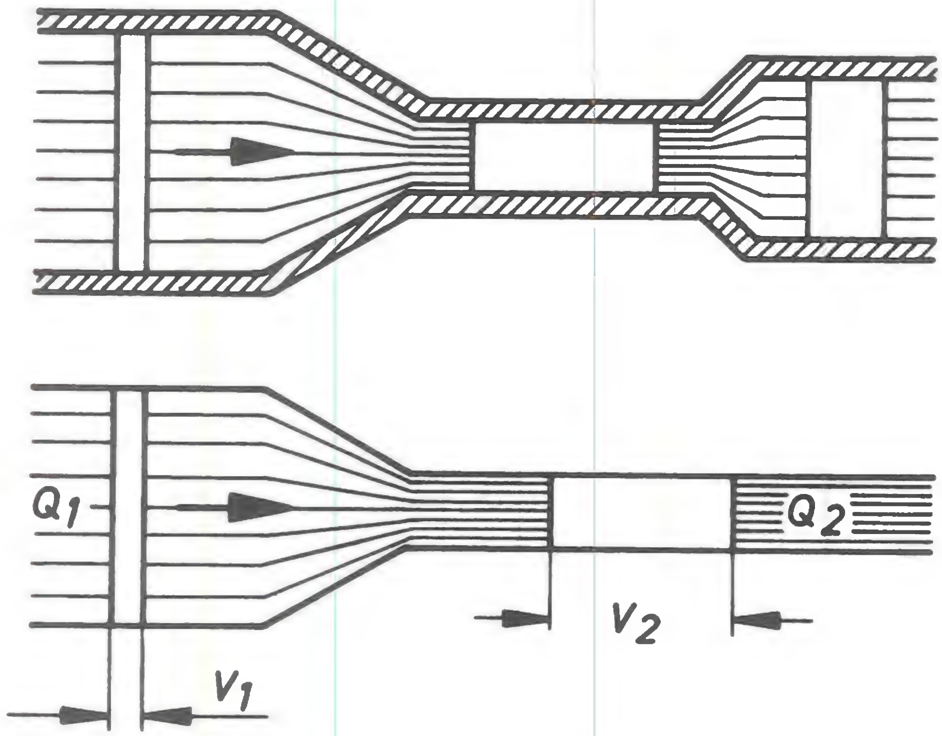
With pressure transmission, pressure acts oppositely to the surfaces.

Basic Principles

Hydrodynamics (mechanics of moving fluid)

Flow Law

fig. 5



If fluid flows through a pipe with different diameters, the equal volumes flow in equal time (fig. 5).

The speed of the flow volume changes.

Flow volume $Q = \frac{V}{t}$

Q = flow volume in $\frac{\text{litres}}{\text{min}}$

V = volume in litres

t = time in min.

A = section area

s = path (length)

Volume $V = A \cdot s$

used in $Q = \frac{A \cdot s}{t}$

The path per time t = speed v ($v = \frac{s}{t}$)

and there results from this with $Q = A \cdot v$

the continuity equation $A_1 . v_1 = A_2 \cdot v_2$ $Q_1 = Q_2$

Energy Law (Bernoulli's equation)

The energy law applied to a flowing fluid says that the total energy of a flow of fluid does not change, as long as energy is not supplied from the outside or drained to the outside.

If you do not take into consideration the types of energy which do not change during flow, the total energy is made up of:

potential energy — **potential energy** related to the height of the head of liquid

pressure energy of static pressure

and kinetic energy — **movement energy** (pressure head) related to flow speed.

Basic Principles

Bernoulli's equation

$$g \cdot h + \frac{p}{\rho} + \frac{v^2}{2} = \text{constant}$$

related to pressure energy, this means:

$$p_{total} = p_{st} + \rho \cdot g \cdot h + \frac{\rho}{2} \cdot v^2$$

p_{st}	= static pressure
$\rho \cdot g \cdot h$	= pressure via the height of the head of fluid
$\frac{\rho}{2} \cdot v^2$	= pressure head

If you now look at the continuity equation and the energy equation, the following situation results:

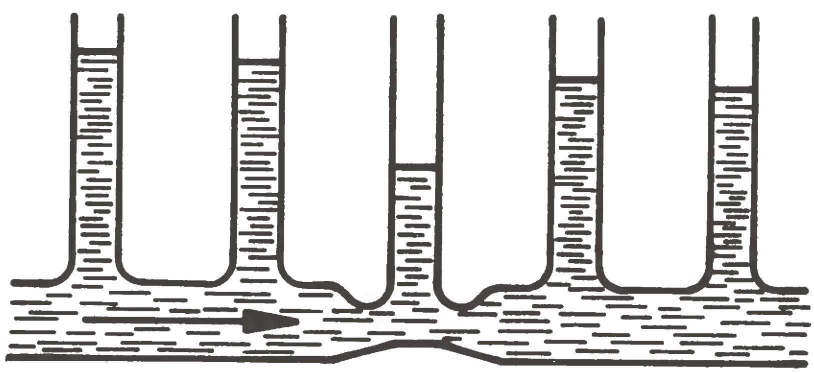
If speed increases as the diameter decreases, the movement energy increases.

Since total energy remains constant, potential energy or pressure energy, or both, must change, i.e. decrease, when the diameter decreases.

However, decrease in diameter causes a scarcely noticeable change in potential energy.

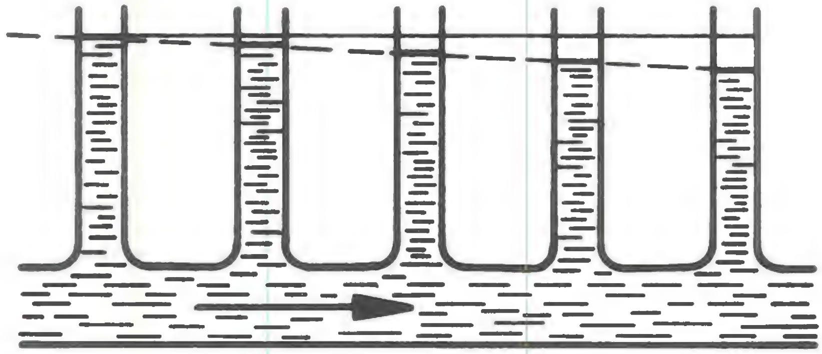
Static pressure thus changes in relation to normal pressure, i.e. in relation to the flow speed (fig. 6).

fig. 6



The height of the heads of liquid is a dimension for the pressure at that position.

fig. 7



In a hydraulic unit, pressure energy (static pressure) is the main decisive factor, since the fluid level and flow speed are too low.

Loss of Energy by Friction

If the fluid is still (no fluid movement), then pressure in front of, in and behind a throttle position or generally in a line are equal.

If fluid is flowing through the system, heat is created by friction. Thus part of the energy is lost as heat energy, which means loss of pressure (fig. 7).

Hydraulic energy cannot be transferred without loss. The amount of friction loss is related to, for example:

length of the pipe

roughness of the pipe wall

number of pipe bends

pipe diameter

flow speed

Flow Configurations

The flow configuration and thus the friction losses are related to the two last mentioned points, pipe diameter and flow speed.

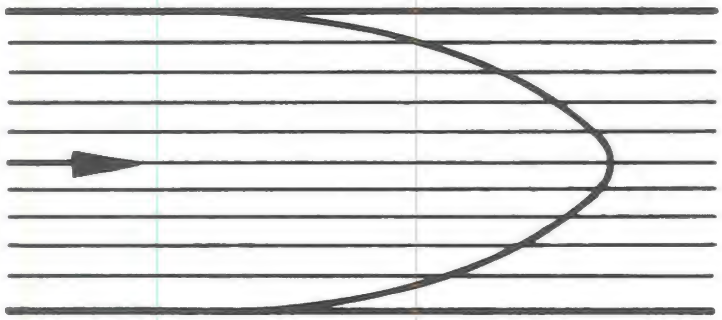
a) Laminar Flow

In laminar flow, the individual fluid particles move up to certain speeds in uniform layers alongside each other. They scarcely disturb or influence each other (fig. 8).

b) Turbulent Flow

If the flow speed increases when the pipe diameter remains the same, the flow characteristic changes after a certain speed (critical speed). Flow becomes whirling and turbulent. The individual particles no longer move in one direction in an orderly fashion, but influence and hinder each other.

fig. 8



Laminar Flow

Basic Principles

fig. 9

Turbulent Flow

The flow resistance and hydraulic losses increase.

Turbulent flow is therefore not desirable in hydraulic units (fig. 9).

Reynold's Number R_e

The flow can be determined using Reynold's number.

$$\boxed{R_e = \frac{v \cdot d_H}{\nu}}$$

R_e has no dimension

v = flow speed $\left(\frac{m}{s}\right)$

d_H = hydraulic diameter (m)
with circular sections = pipe internal diameter, otherwise calculate

$d_H = 4 \times \frac{A}{U}$

A = section area

U = circumference

ν = kinetic viscosity $\left(\frac{m^2}{s}\right)$

R_e critical $\approx$ 2300

This value is valid for round, technically smooth, straight pipes.

At R_e critical, the flow configuration changes from laminar to turbulant, or vice versa.

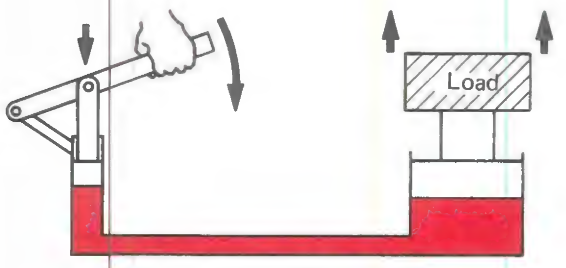
laminar flow R_e $<$ R_e critical

turbulent flow R_e $>$ R_e critical

Basic Principles

Basic Form of a Hydraulic System

Fig. 10



The above shows in principle the basic form of a hydraulic system.

We load the piston of a single piston pump with a certain force. Force divided by piston area results in the pressure which can be achieved in every case ($p = \frac{F}{A}$).

The more we press on the piston, i.e. the greater the force on the piston is, the higher pressure rises. However, it rises only until, related to the cylinder area, it is in a position to overcome the load. ($F = p \cdot A$).

If the load remains constant, pressure does not increase further.
Consequently, it acts according to the resistance, which is opposed to the flow of the fluid.

The load can therefore be moved, if the necessary pressure can be built up.
The speed, at which the load moves, is dependent solely on the volume of fluid which is fed to the cylinder.

With reference to fig. 10, this means that the faster the piston is lowered, the more fluid per time unit is supplied to the cylinder, and the faster the load will lift.
However, in practice, we must enlarge on this system.

We wish to fit devices, with which we can influence, for example:

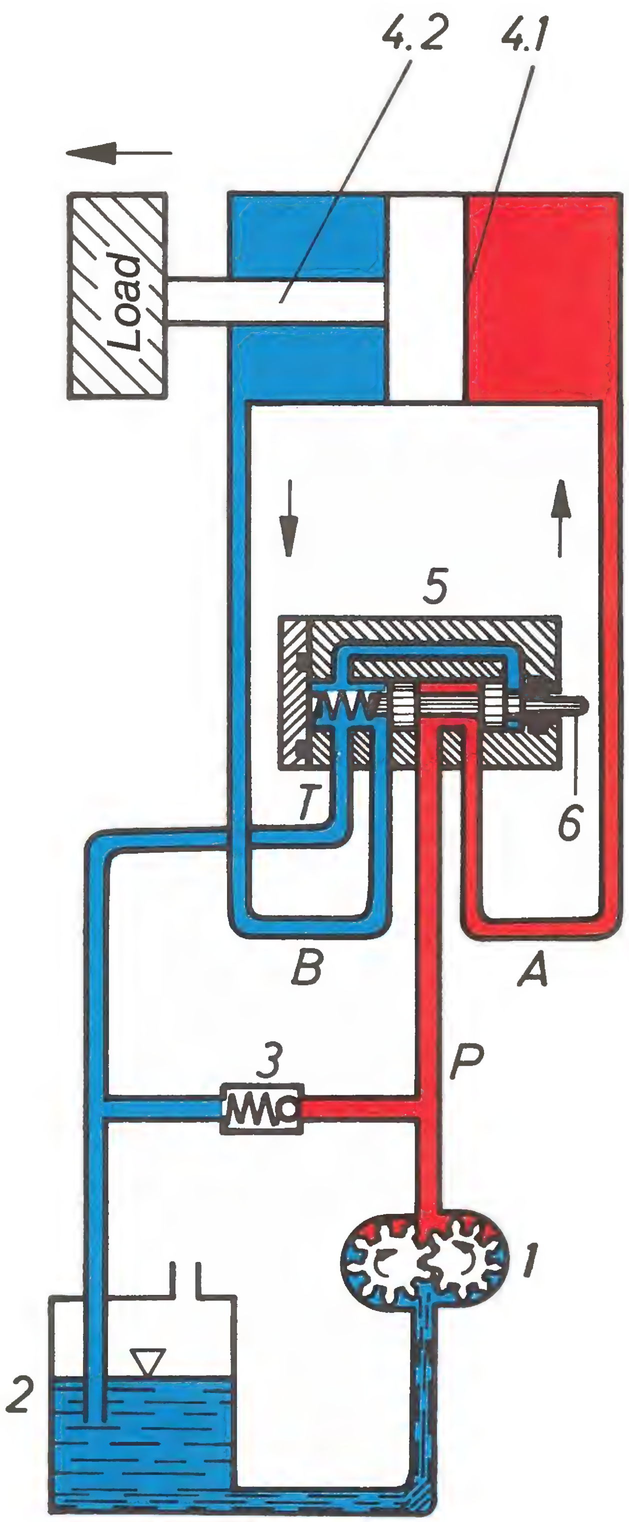
the direction of movement of the cylinder
the speed of movement of the cylinder and
the maximum load of the cylinder.

Also, we shall replace the manually operated piston pump with a continuously driven pump.

To make this easier to understand, we shall show a diagram of a simple hydraulic circuit.

Basic Principles

Fig. 11



Pump 1 is driven by a motor (electric motor or internal combustion engine) (fig. 11).

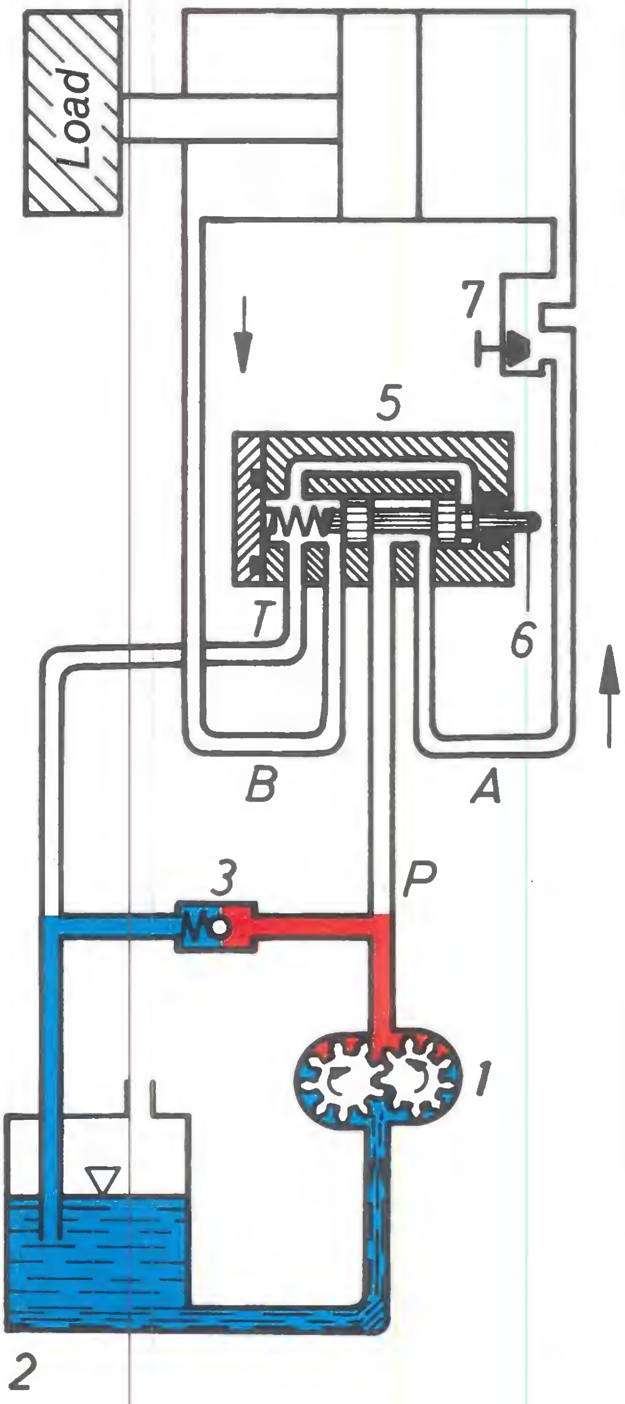
It sucks fluid from tank 2 and pushes it into the following line system with various devices up to cylinder 4 (or also hydraulic motor). So long as there is no resistance to the flow, the fluid is merely pushed further.

Cylinder 4 at the end of the line, for example, represents such resistance to the flow.

Pressure therefore increases until it can overcome this resistance, i.e. until the cylinder moves.

Basic Principles

Fig. 12



The maximum pressure must be limited, so that the system is protected from too high a load (that means at the same time from too high a pressure).

This is achieved using pressure relief valve 3. A spring as mechanical force, presses a ball on the seat. Pressure in the pipe affects the ball surface.

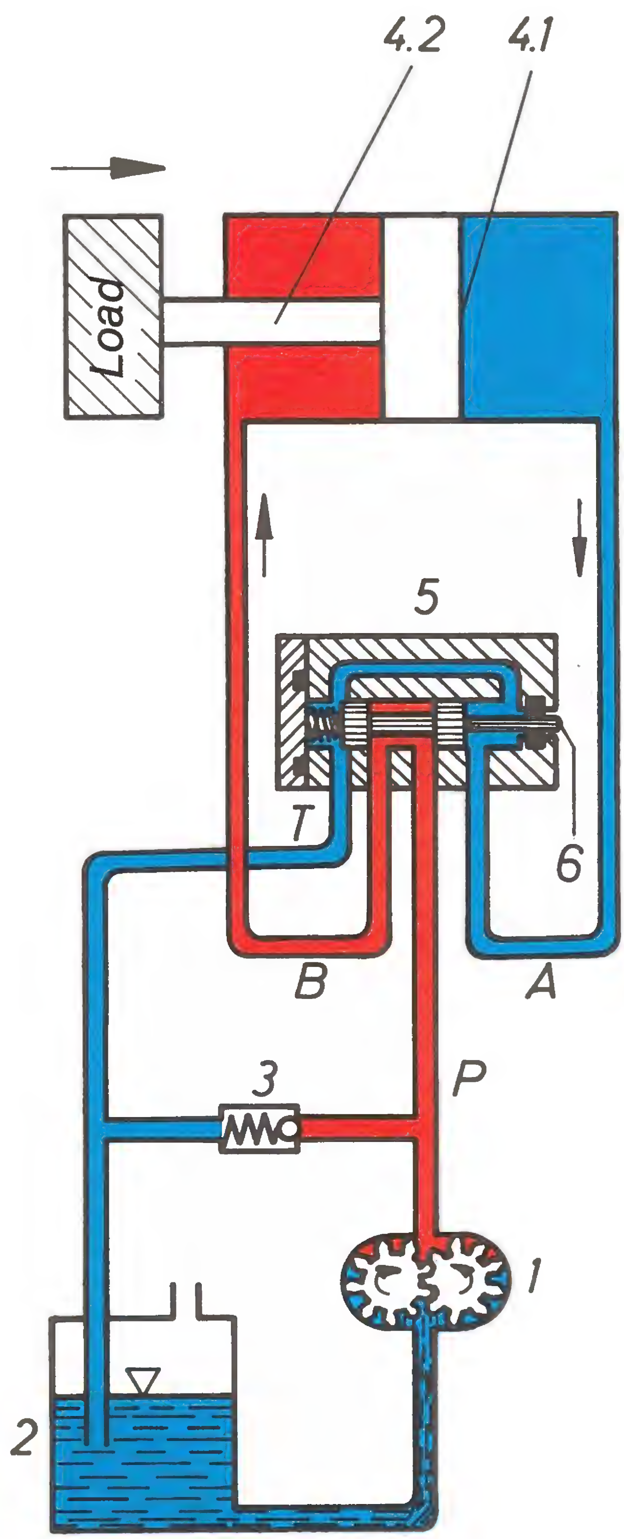
According to the equation already mentioned, $F = p \cdot A$, the ball opens when the force from pressure X area exceeds the spring force.

Pressure now no longer rises.

The full flow delivered by the pump flows back to tank via valve 3 (fig. 12).

Basic Principles

Fig. 13



Control valve 5 (directional control valve) determines whether piston 4.1 with piston rod 4.2 travels into or out of the cylinder (fig. 13).

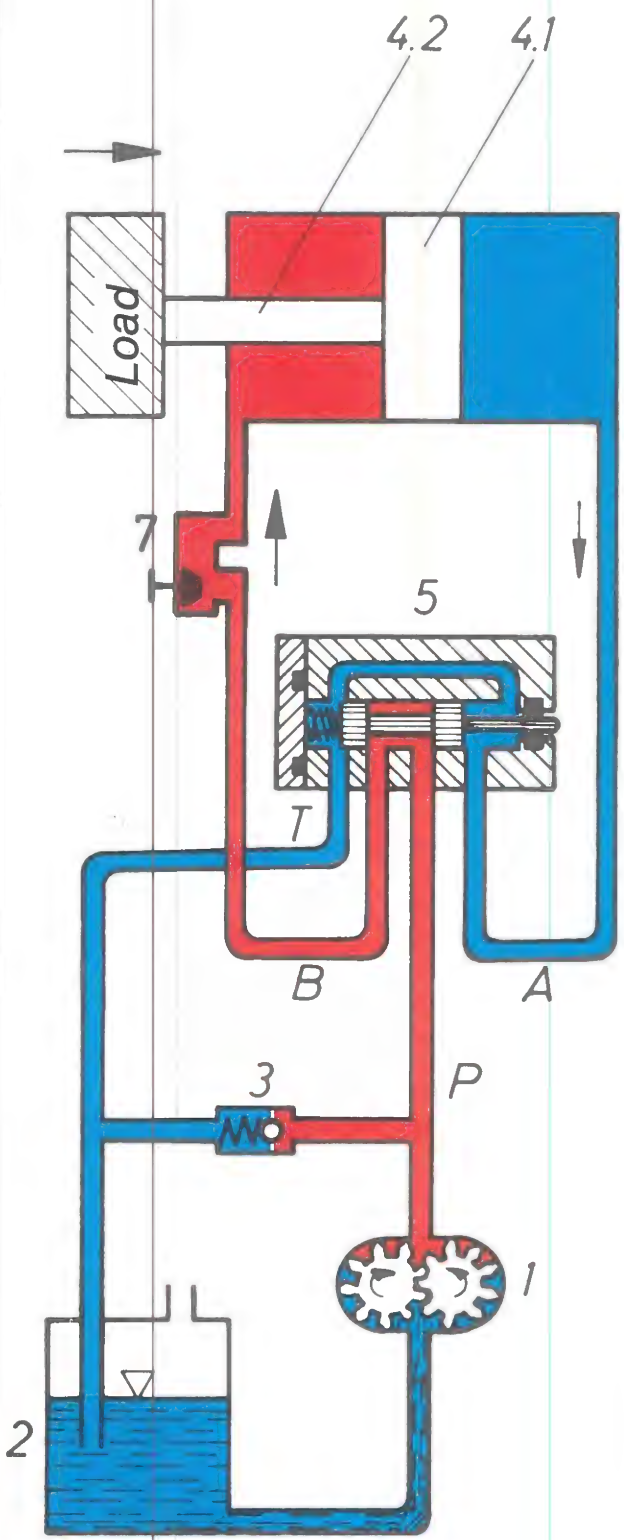
In fig. 11, the fluid at valve 5 has flowed from line connection P to A to the cylinder.

Pushing spool 6 in the control valve provides a connection from P to B. The fluid now flows from the pump via the valve into the other side of the cylinder.

Piston rod 4.2 travels inwards. The load now moves in the other direction.

The fluid from the opposite chamber is pushed back to tank via control valve 5 from A to T.

Fig. 14



The volume of fluid flowing to or from the cylinder must be changed, if you wish to influence not only the direction of movement and the force, but also the speed of movement of the load.

This is achieved, for example, with a throttle valve 7 (fig. 14).

By changing the flow section (decrease related to the line section), less fluid per time unit (in our example here) flows to the cylinder.

(Note: the conditions at a throttle position are described in detail in the section "Flow Control Valves".)

The load moves slower.

The excess fluid, which is now delivered by the pump, must drain via the pressure relief valve.

Referring to the pressure conditions in the system, this means that:

the pressure between the pump and the throttle is the maximum pressure set at the pressure relief valve.

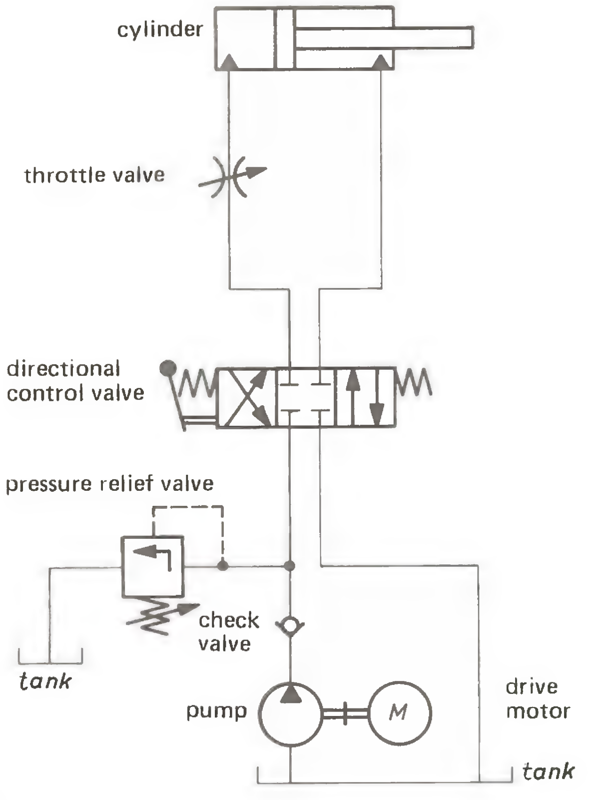
the pressure between the thottle and the cylinder depends on the load.

Basic Principles

Basic Diagram of a Hydraulic Circuit

In practice, a hydraulic circuit is not shown graphically as in figs. 11 to 14.

fig. 15



Symbols are used in place of simplified sectional drawings.

The graphic representation of a hydraulic circuit with these symbols is called a circuit diagram. The diagram and the meaning of the individual devices and function are shown in the DIN–ISO 1219 standard.

The relevant symbols will be shown in connection with component descriptions.

Notes

Examples of Application from Various Sectors of Industry

In order to give you a full impression of the application ranges of hydraulics, we have divided these into 5 sectors as follows:

1. Industrial Hydraulics	Plastic Machines Presses Heavy Machinery Machine Tools
2. Hydraulics in Steelworks, Civil Engineering and Generating Stations	Lock Gates and Dams Bridge Operating Equipment Mining Machinery Turbines Nuclear Power Stations
3. Mobile Hydraulics	Excavators and Cranes Constructional and Agricultural Machinery Automobile Construction
4. Hydraulics in Special Technical Applications	Telescopes, Antenna Operation, Recording Buoys, Landing Gear and Rudder Control of Aircraft Special Machinery
5. Hydraulics for Marine Applications	Rudder Controls Deck Cranes Bow Doors Bulkhead Valves

Naturally, this summary does not include all possibilities of application, since the variety of hydraulically operated machines is too great.

However, it can be seen that hydraulics have today made headway in almost all sectors of industry.

A few photographs are shown on the following pages to supplement this summary.

Examples of Application from Various Sectors of Industry

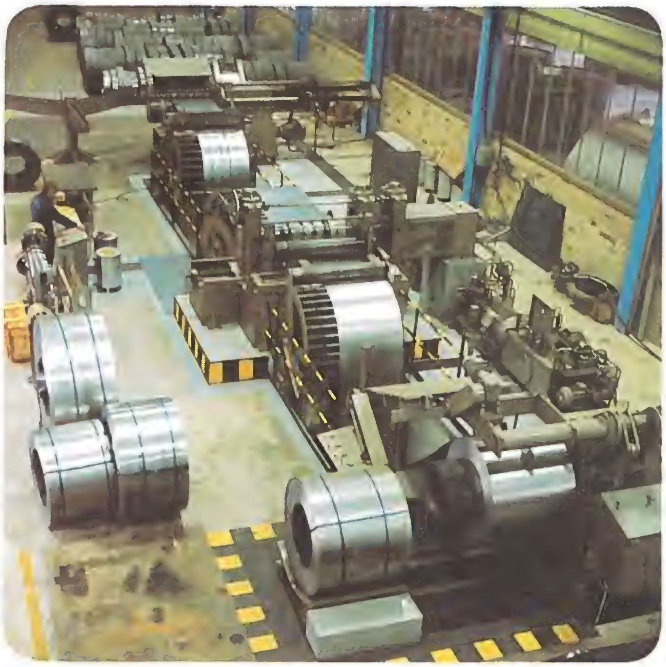
Hydraulics for Steelworks

Today it is difficult to find a steelworks or rolling mill, which is not fitted with extensive hydraulic equipment.

The high production level on these valuable machines could not be considered without these "hydraulic muscles", which react to a push button.

The following, for example, are hydraulically operated:

- elevating platform
- rocking lever
- feeding and output transport table
- tilting table
- channel
- adjustment of rolls
- sorting and transporting devices
- movements on the cooling line.



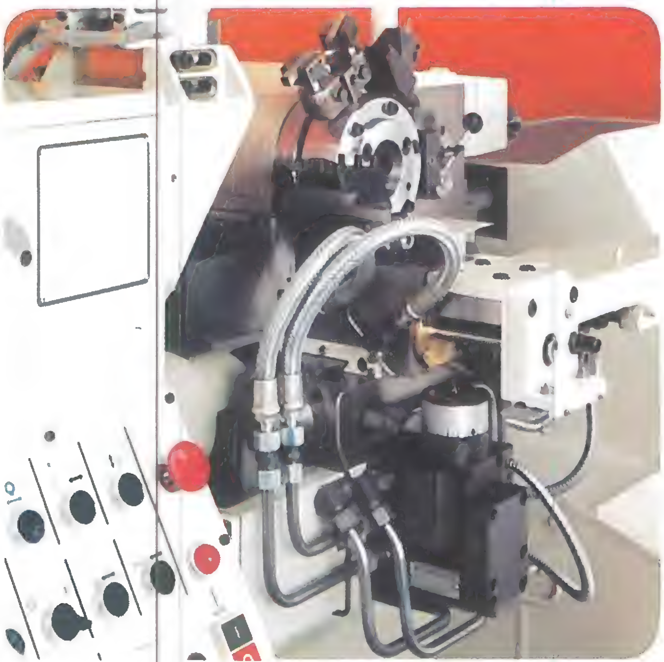
Fully automatic strip cutting line for stamping core plate

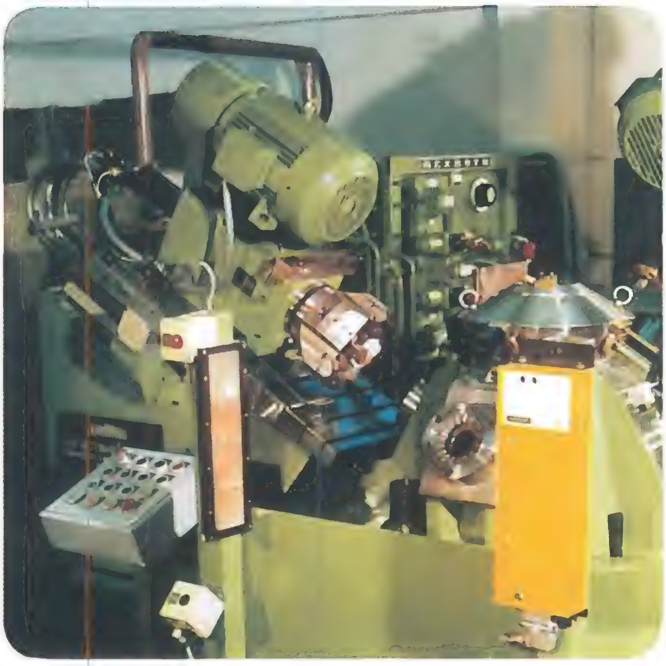
The main drive comprises electrically variable axial piston pumps and pressure controlled hydraulic motors in closed circuit.

The linear strip speed is up to 100 m/min.

Examples of Application from Various Sectors of Industry

Hydraulics for Machine Tool Applications





Precision is a positive characteristic of present-day hydraulic controls. Even for widely used production equipment, the optimum solution can be provided for each particular function with a tried and tested system.

Specialised hydraulic equipment makes this positive type of control simple and economic, even for complex operations, and the design enables it to be incorporated into the overall machine concept more easily than with other systems.

Being able to integrate various control groups into standardised modules on a machine is a great step into the future for the further development of metal-cutting machine tools, which one cannot contemplate today without hydraulics.

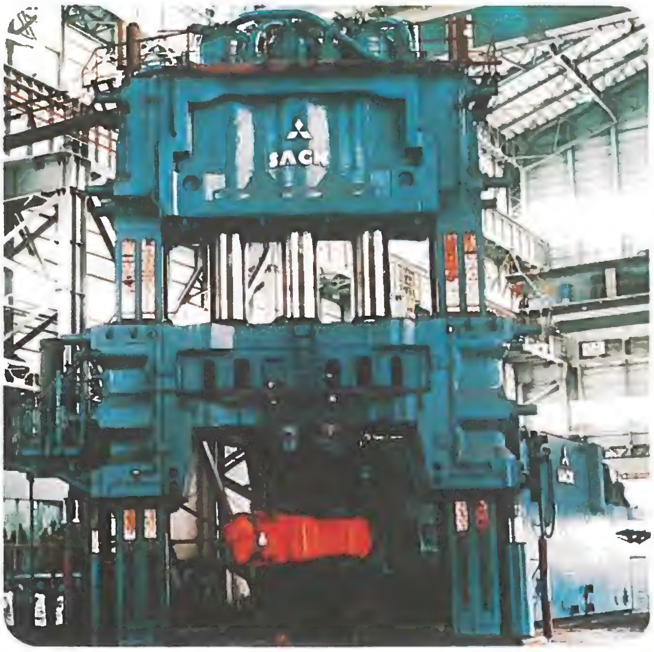
Hydro-mechanical tracing, as seen on this automatic lathe, is an excellent example of precision hydraulics.

Four station/2 directional surface grinding machine for machining cold extrusion moulded gears.

Two control units, the workpiece clamp and holder and the circular table are all hydraulically operated.

Examples of Application from Various Sectors of Industry

Hydraulics for Press Applications



Forging press with 120 MN (12 000 Mp) force.
The use of hydraulics is the easy solution.

Safety of personnel and machinery is always guaranteed, in spite of the impressive production capacity of such machinery being fully utilised.

The favourable power density of hydraulics is displayed in such applications. Hydraulic pumps and valves can be fitted more easily into a small space, than is the case with mechanical systems.

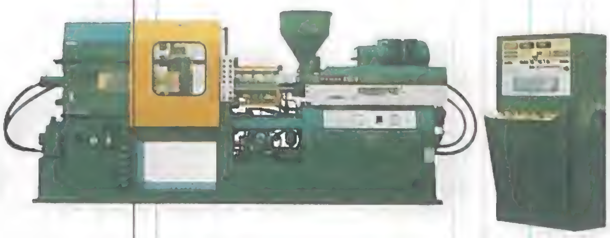
Baling press for processing scrap automobile bodies into compact metal bales.

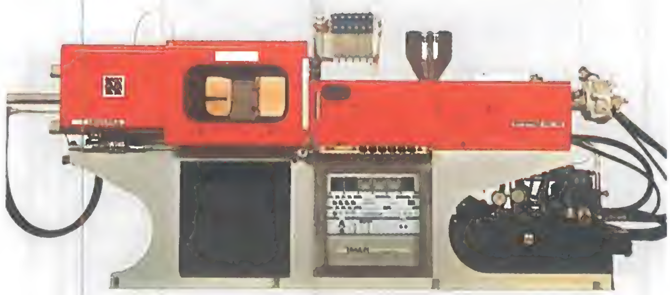
The process of operation is completely hydraulic and comprises 3 stages:

1. loading and cutting
2. vertical pressure
3. horizontal pressing and ejection of bales on to a conveyor.

Examples of Application from Various Sectors of Industry

Hydraulics for Pressure and Injection Moulding Machines



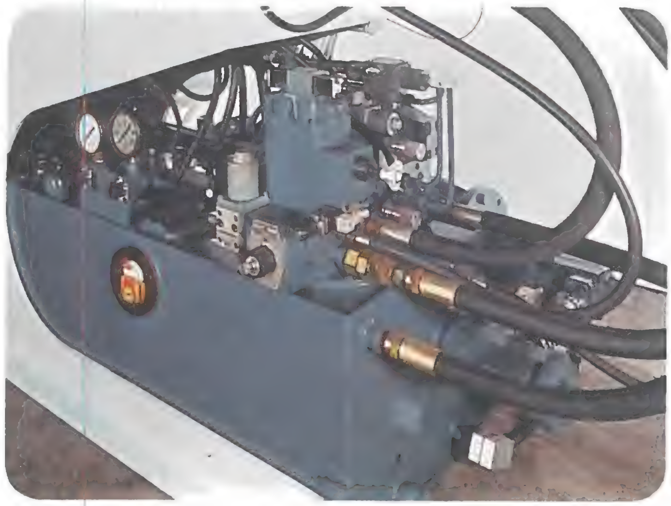


When using the telephone, lifting a bottle of washing-up liquid, or watching a child playing with a plastic toy, one seldom thinks about the machinery which produces all our modern household commodities.

There are, however, thousands of plastic processing machines all over the world, which are fitted with elaborate hydraulic power transmission systems for movement of the platten, holding the workpiece or for stepless adjustment of closing forces and speeds.

On this injection moulding machine, closing forces of 20 – 280 tons are required, depending on the size of the machine.

Uses of the machine vary from plastic cups to gears, depending on the machine tool used.



Examples of Application from Various Sectors of Industry

Hydraulics for Mobile Machinery

The connection between the drive source and the power output should always be as short as possible, and have a minimum loss.

In construction machinery of the latest design, hydraulic energy transmission solves this old mechanical problem in a simple and much more effective way, since the fluid energy transmission can be carried out by means of flexible hoses to the actuators for the moving parts.

Direct power flow is best demonstrated on excavators. The robot type movements illustrate the hydraulic amplification of the manual force of an individual and follow exactly the control of the operator.

Hydraulics make full use of their great advantage of infinite variability of traction and speed in applications in hydrostatic transmissions of construction machinery.

The variety of machines used in this sector, and thus also the variety of possibilities for application for hydraulics, is shown by the above vehicle for bridge inspection.

Gondolas for inspection and maintenance personnel, as well as the footbridge, are positioned by means of hinged brackets.

The telescopic footbridge will stretch to a length of 20 m.

Not only force, but also safety and accuracy of operation are required here.

Examples of Application from Various Sectors of Industry

Hydraulics for Handling Equipment

As in all cases, time is a very important factor here. A load must be discharged quickly and freight then loaded exactly and safely to its next means of transport.

On this handling equipment, hydraulics accomplish the manoeuverability of the scraper at the bending and lifting gear or at the rotating mechanism of the equipment.

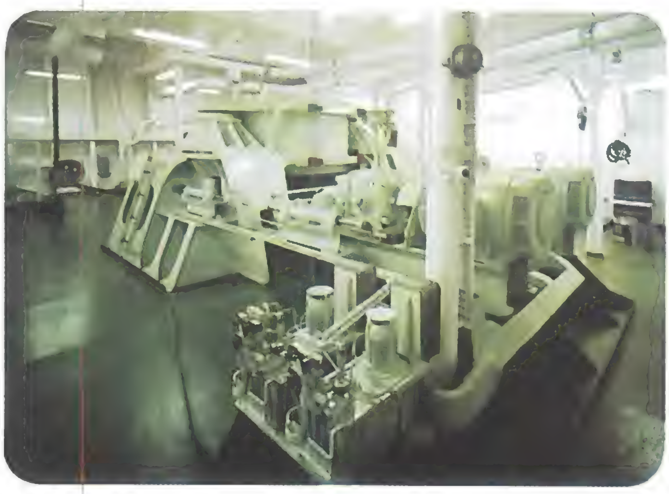
However, hydraulics are used not only on this type of equipment, but also on the equipment used for loading our holiday baggage at the airport.

Hydraulics for Marine Applications

A cost-saving rationalisation programme using hydraulics has brought about many important changes and improvements in the design and fitting of ships, both above and below deck.

Hydraulically driven and adjustable stern thrusters guarantee manoeuverability of giant ocean-going vessels in harbour. Discharging of tankers is controlled hydraulically in the extensive pipe network on these ships. A further example is the operation of nets on fishing boats — this heavy work is made easier and can be carried out in a shorter time if hydraulically operated winches are used.

An example of this is the rotating wing rudder machine shown below.



Examples of Application from Various Sectors of Industry

Hydraulics for Civil Engineering

The very large and urgent task of developing coastal protection and waterways can be carried out very economically by using hydraulic control and regulating techniques.

In order to contain enormous natural forces, very large hydraulic cylinders move shields against the flood tides so that the inhabitants of the endangered coastal region can live in safety.

The increase in shipping using Europe's waterways has meant that there will have to be a rapid extension of these waterways to accommodate greater freight capacities.

Only by using hydraulics in a series of lock-gates, is it possible to ferry goods quickly and safely, both upstream and downstream.

A further impressive example is the ship lifting channel in the Elbe side canal, with which a height difference of 38 metres is achieved.

In this case, hydraulics are fitted to operate the various cut-off walls, the lifting platform and for four buffer devices.

Opening and closing of drawbridges and lock gates is also included in this field of application.

Examples of Application from Various Sectors of Industry

Hydraulics for Special Applications

Many developments from Rexroth have opened up new applications for hydraulic drives and controls.

Hydraulic equipment is a matter of course on handling and lifting equipment, on mining machinery, as well as on drilling rigs for oil drilling in the North Sea. Agricultural machinery and tractors, fitted with hydraulics, perform important functions in the farming industry. Everything profits from this rationalised power transmission.

The complicated controls of huge aerials, by means of which important news is transmitted from continent to continent, require a very high degree of precision.

This is possible using hydraulics as drive, control and regulating components. The functional safety of hydraulics is guaranteed.

Functional safety is required for the application of hydraulics in this measuring buoy in the North Sea.

The measuring buoy is one component in the installation of a measuring network in the North Sea and Baltic Sea, to collect important oceanographic and meteorological data.

Winch drive and control for the measuring cable are hydraulically operated.

Programme Summary

Pumps

Fixed Displacement

Gear Pumps
Size 3 . . . 100
3.5 . . . 100 cc/rev.
. . . 250 bar

Vane Pumps
Size 10 . . . 100
10 . . . 100 cc/rev.
. . . 175 bar

Radial Piston Pumps
Size 0.6 . . . 19.4
0.6 . . . 19.4 cc/rev.
. . . 700 bar

Axial Piston Pumps
Size 10 . . . 2 000
10 . . . 2 000 cc/rev.
. . . 400 bar

Variable Displacement

Vane Pump
Type V3
Size 12 . . . 63
8.5 . . . 47 cc/rev.
. . . 100 bar
Type V4
Size 20 . . . 125
20 . . . 125 cc/rev.
. . . 160 bar

Axial Piston Pumps
Size 10 . . . 2 000
10 . . . 2 000 cc/rev.
. . . 400 bar

Programme Summary

Motors

Fixed
Displacement

Gear Motors

Size 3.5 . . . 100 cc/rev.
. . . 250 bar
. . . 207 Nm

Radial Piston Motors

Size 33 . . . 5 275 cc/rev.
. . . 315 bar
. . . 24 250 Nm

Axial Piston Motors

Size 10 . . . 2 000 cc/rev.
. . . 400 bar
. . . 11 110 Nm
(at 350 bar Δp)

Slow-Speed Motors

Size 47 . . . 1 000 cc/rev.
. . . 350 bar
. . . 4 380 Nm

Variable
Displacement

Axial Piston Motors

Size 28 . . . 468 cc/rev.
. . . 400 bar
. . . 2 380 Nm
(at 320 bar Δp)

Hydraulic Pumps and Hydraulic Motors

Hydraulic pumps and motors are hydrostatic components.

The conversion of mechanical torque by means of operating pressure and stroke volume, or vice versa, is the same on all hydrostatic machines.
For torque, this can be seen from the basic formula (without degree of efficiency).

$$M = \frac{\Delta p \cdot V_h}{2 \cdot \pi}$$

Pump M = drive torque
Motor M = driven torque (without η)
Δp = pressure drop between:
outlet and inlet of the pump
inlet and outlet of the motor
V_h = geometric stroke volume

To achieve this conversion, there are various possibilities from the design point of view.

Fig. 1 shows the principle possibilities:

1) general
volume formation
2) gear unit
3) radial piston unit
4) vane unit
5) axial piston unit with bent axis
6) axial piston unit in swashplate design

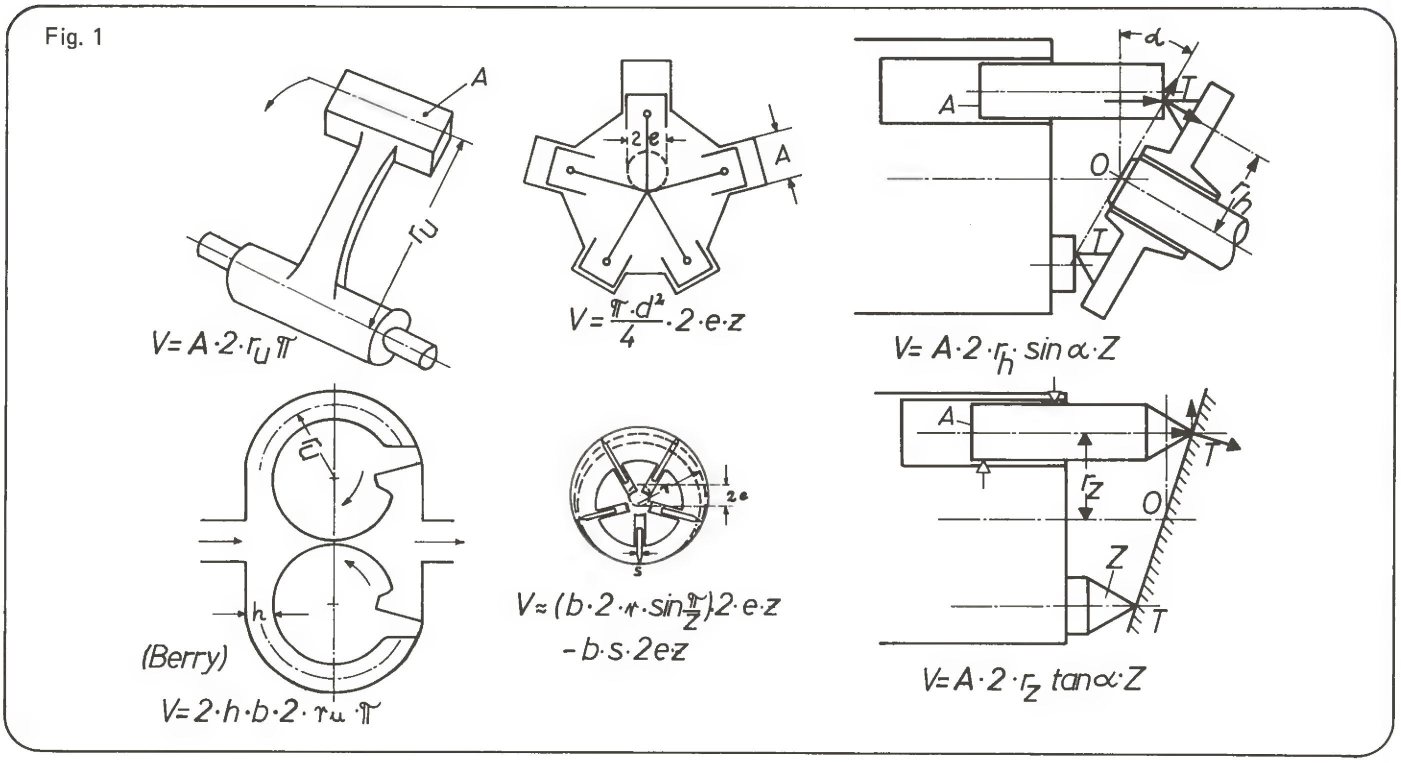
As the fluid in these units is desplaced , they are called displacement units.

They can be sub-divided into 5 basic types:

a) gear pump/gear motor
b) vane pump/vane motor
c) radial piston pump/radial piston motor
d) axial piston pump/axial piston motor
e) screw spindle pump

The advantage of hydrostatic power transmission over other methods of transmission lies partly in the relatively high power density. The power density in this case means operating pressure.

Volume formation in hydrostatic machines.



Hydraulic Pumps and Hydraulic Motors

Apart from the various designs, a differentiation is made between:

- Fixed displacement pumps, fixed displacement motors
  The stroke volume cannot be changed.
- Variable displacement pumps, variable displacement motors
  The stroke volume can be changed.

Hydraulic Pumps

Function:

In hydraulics, pumps serve to create a fluid flow (to displace a volume of fluid) and to allocate the necessary forces to it as required.

The pump sucks fluid (generally from a tank) and delivers it to the pump outlet.

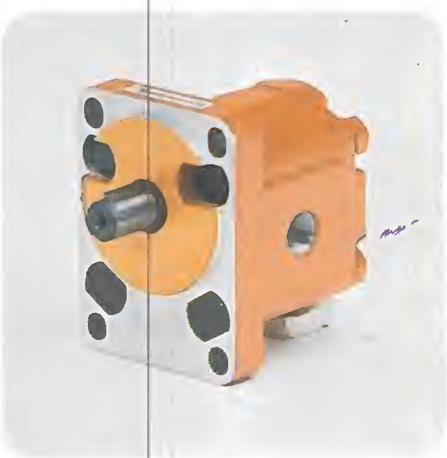
From there, fluid enters the system and reaches the user by means of the individual control elements. The user offers resistance to the fluid, e.g. the piston of a stroke cylinder under load.

According to this resistance, pressure builds up in the fluid until it is high enough to overcome the resistance forces.

The pressure in a hydraulic system is not created by the hydraulic pump, but builds up. This therefore occurs in relation to the resistances, which are opposed to the flow.

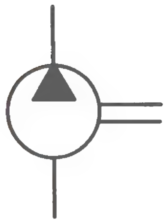
The head of fluid could also be seen as a fluid connecting rod, to which the necessary forces are allocated by the pump.

Gear pumps



Gear Pump with external gear, type G

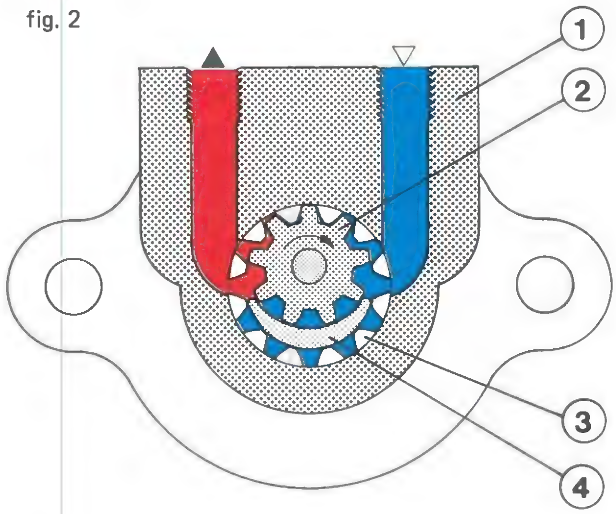
Symbol



Gear pumps are fixed displacement pumps.

Gear pump with internal gearing (fig. 2)

fig. 2



It comprises mainly a housing (1), in which a pair of gears run with such low axial and radial play, that the unit is practically oil-tight.

The suction side (blue) is connected to the tank. The pressure side (red) is connected to the hydraulic system.

The inner gear 2 is driven in the direction of the arrow, and takes external gear 3 along with it in the same direction.

The rotary movement causes the gears to separate, so that the gear spaces are free.

The negative pressure caused by this and the atmospheric pressure on the fluid level in the tank cause fluid to run from the tank to the pump. One generally says "the pump sucks".

The fluid fills the gear spaces, which form closed chambers with the housing and the crescent 4, during further movement, and is pushed to the pressure side (red).

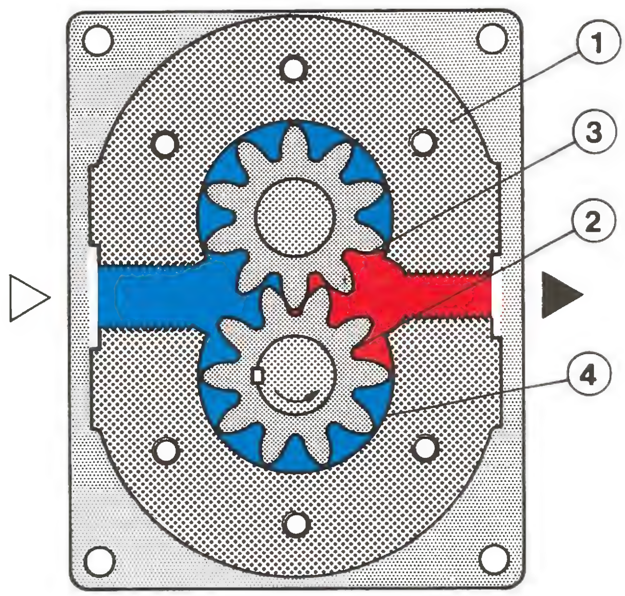
The gears then interlock once more and push the fluid from the gear chambers.

Hydraulic Pumps and Hydraulic Motors

The gears which would come into contact with one another prevent return flow from the pressure chamber to the suction chamber.

Gear Pump with External Gearing (fig. 3)

fig. 3



In this case, 2 external gears would come into contact with one another. Gear 2 is driven in the direction of the arrow, and causes gear 3 to move with it in the opposite direction. The suction process is identical to that previously described for the internally geared pump.

The fluid in gear chambers 4 is pushed round the outside and out of the gear spaces on the pressure side (red).

It can easily be seen from the sectional drawing that the gears close the spaces before these are completely empty.

Without unloading in the remaining chambers, very high pressures would occur, which would result in hard pulsating running of the pump.

For this reason, unloading bores are arranged at this position on the side of the bearing blocks. The so-called "compressed fluid" is thus fed into the pressure chamber.

A further point worthy of note is the side tolerance play between gears 5 and bearing blocks 6. fig. 4

If tolerance play too great:	low friction high leakage
If tolerance play too low:	high friction low leakage

If the tolerance play is designed as a fixed aperature, leakage increases along with wear. The volumetric loss also increases with increasing operating pressure.

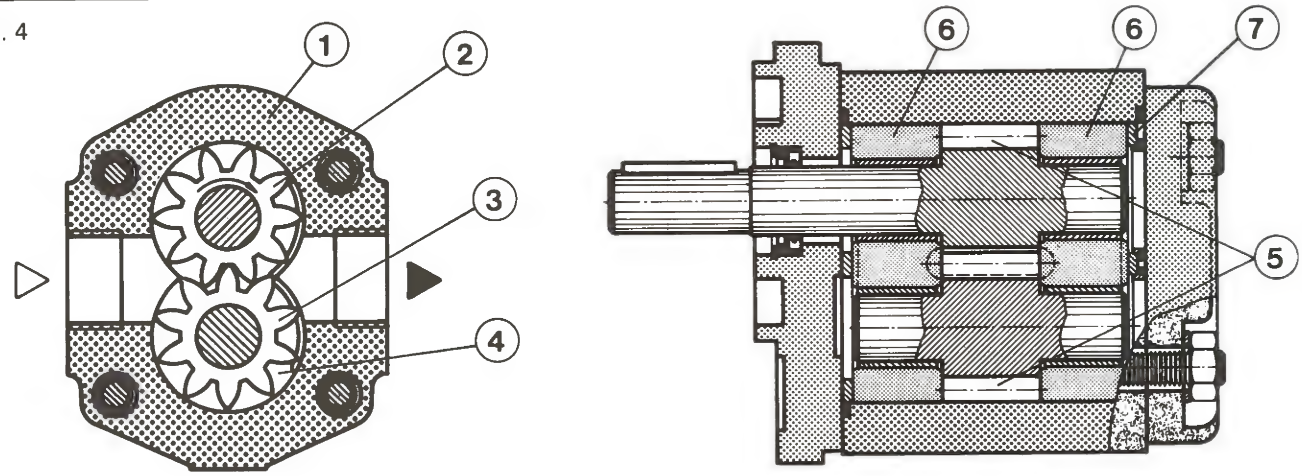
This pump design incorporates a hydrostatic bearing balance. The bearing blocks are pushed on to the gears by cams 7, which are affected by system pressure.

Tolerance play therefore adjusts itself according to the system pressure. This results in a high degree of efficiency, independent of speed and pressure.

Important technical details:

displacement volume:	3.5 – 100 cc/rev.
operating pressure:	up to 250 bar

Fig. 4



Hydraulic Pumps and Hydraulic Motors

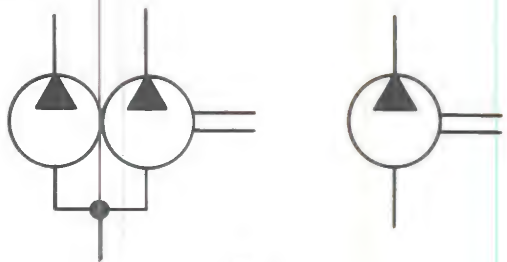
Vane Pumps

Vane Pumps with fixed displacement volume

Vane pumps type V2

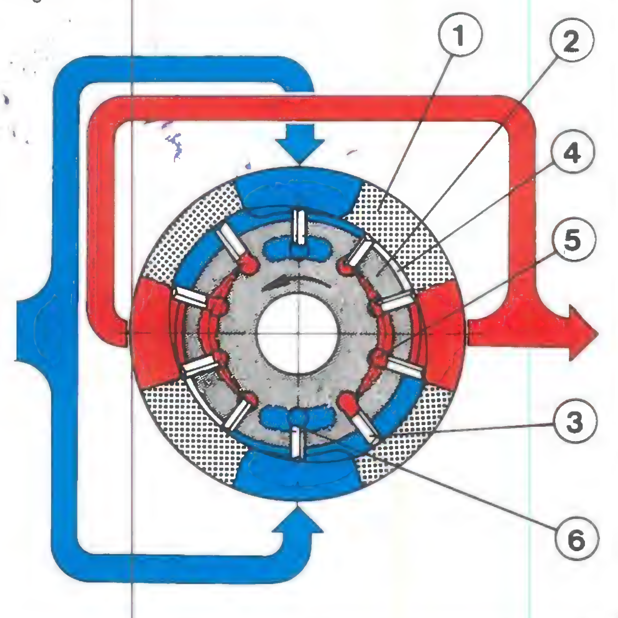
on the right: single pump
on the left: double pump

Symbol



The schematic sectional diagram shows the design of this pump principle (fig. 5)

Fig. 5



The vane pump comprises mainly the housing, cam 1 and rotor 2 with the vanes 3.

Cam 1 has an internal running surface in double eccentric design. The rotor is the drive part. On its circumference, two vanes 3 (double vanes), which can be pushed against each other, are fitted in radially arranged grooves.

When the rotor is turned, the centrifugal force and the system pressure behind the vanes push the radially moveable vanes towards the outside. They lie with their external edge to the internal running area of the cam.

The cells (transporting chambers) are formed by 2 pairs of vanes, the rotor, the cam and the control discs arranged on the side.

Supply (suction side, blue) and drain (pressure side, red) of the fluid take place by means of the control discs (not shown).

To make things easier to understand, external supply and drain are shown in the drawing (fig. 5).

For flow delivery, the rotor is moved in the direction of the arrow. Near the suction line (above and below), the vanes 4 are still too small. If the rotor is turned further, the vanes increase and fill up with oil. When these cells have reached their maximum size, (largest distance from the internal running space to the centre point of the rotor), they are separated from the suction side by means of control discs.
They are then connected to the pressure side.

The vanes are pushed into the grooves by the form of the cam curve. The vane volume decreases once more. The fluid is thus pushed to the pressure port.

As the cam curve is designed as double eccentric, each vane is involved in the delivery process twice per revolution.

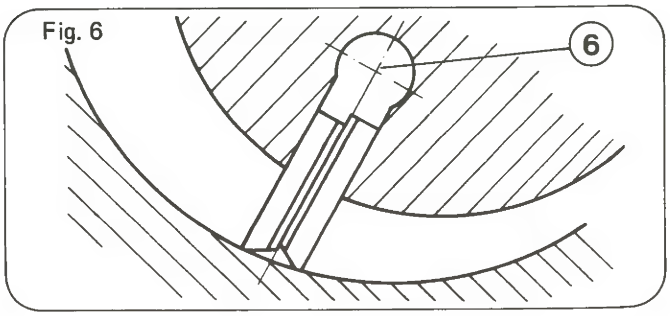
Hydraulic Pumps and Hydraulic Motors

At the same time, two suction chambers and two pressure chambers lie opposite each other, whereby the drive shaft is hydraulically unloaded.

The pressure is applied to the back of the vanes 5.

Better sealing is thus achieved in addition to the double seal lands.

Fig. 6



However, as friction may not be too great, both vanes in a rotor groove have chamfers opposite each other (fig. 6).

The chamfers on the vanes cause a pressure balance between the running and return side. The rolling surface of the vanes remains as contact surface for the pressure. Higher contact pressure is not necessary on the suction side. The backs of the vanes 6 are thus unloaded to tank.

By use of 2 pump elements (rotor, cam, control discs on one shaft in a housing, the double pump with one suction connection and two pressure ports not related to each other is obtained, as shown in the photograph.

Important technical data:
Single pump

Displacement volume:	10 — 100 cc/rev.
Operating pressure:	up to 175 bar

Hydraulic Pumps and Hydraulic Motors

Vane pumps with variable displacement and pressure control

Vane pumps type V3

on the left: subplate mounting
on the right: threaded connections

Symbol

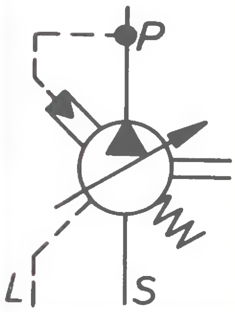
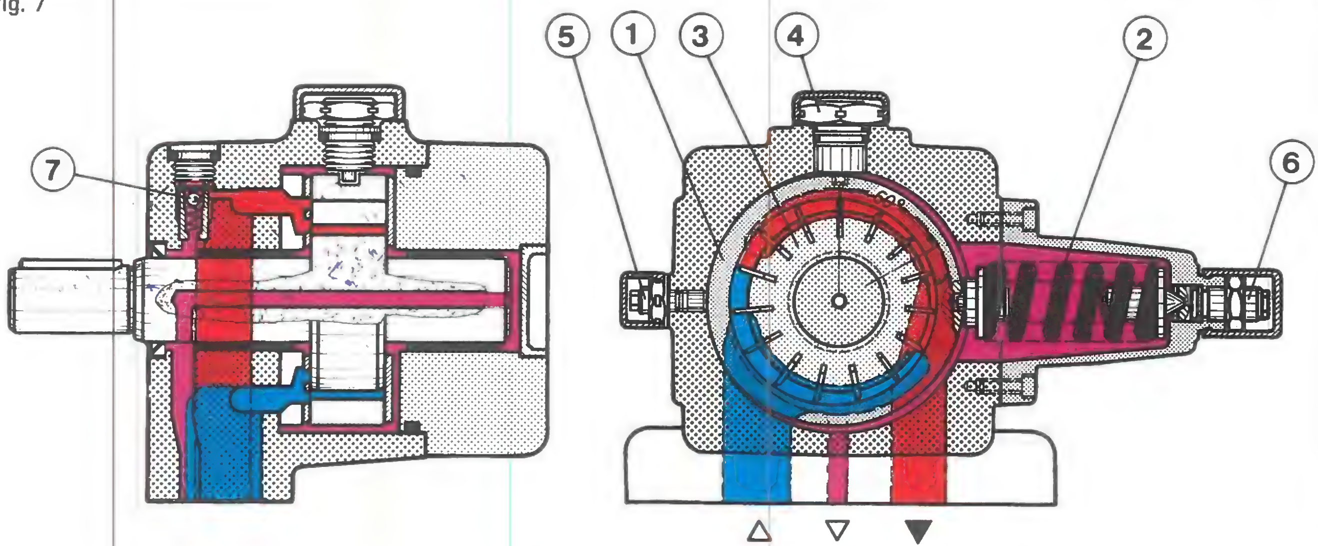


fig. 7

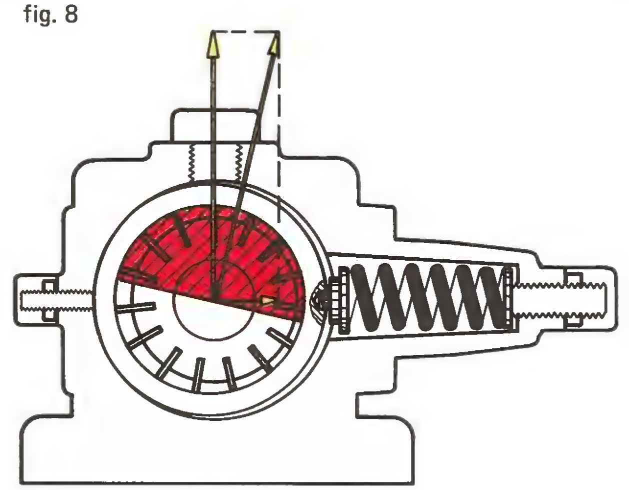


Hydraulic Pumps and Hydraulic Motors

With this type of pump, the displacement volume can be adjusted at the set maximum operating pressure.
The delivery process follows the principle as described for fixed displacement pump V2.
In this case, however, the cam is a circular concentric ring. A spring 2 pushes the cam into its eccentric outlet position towards the rotor.
The maximum eccentricity and thus the maximum displacement volume can be set by means of the screw 5. The spring force can also be adjusted by means of adjustment screw 6.
There is tangential adjustment of the cam by means of the height adjustment screw 4.

The pressure which builds up due to working resistance, (e.g. at the user, cylinder with load) affects the internal running surface of the cam on the pressure side. This results in a horizontal force component which operates towards the spring (fig. 8).

fig. 8



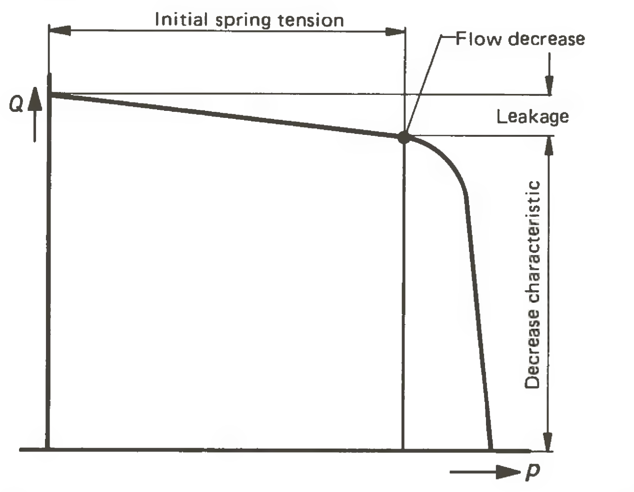
If the pressure force exceeds the set spring force (equivalent to a certain pressure), the cam ring moves from eccentric towards zero position. The eccentricity decreases.

The delivery flow adjusts itself to the level required by the user.

If no fluid is taken by the user and the set pressure is thus reached, the pump regulates flow to almost zero. Operating pressure is maintained, and only the leakage oil replaced. Because of this, loss of power and heating of the fluid is kept to a minimum.

The Q-p-curve shows the behaviour of the pump. (fig. 9)

fig. 9



When the initial spring tension is achieved, the cam ring moves. Flow decreases; pressure is maintained.

The gradient of the performance curve depends on the spring characteristic, whereby the gradient with a certain spring and different pressure varies.

To improve the sensitivity of response, the pump can be fitted with 4 different springs (corresponding to 4 pressure ratings).

Fig. 7 also shows the bleed valve 7 fitted as standard. Commissioning is made easier by automatic bleeding.

The valve poppet is pushed back by the spring; the valve is open. The valve remains open as long as air escapes during start-up. If fluid flows through the valve, the poppet is pushed against the spring and the connection closed hermetically.

Important technical data:

displacement volume:	up to 47 cc/rev. (4 sizes)
operating pressure:	up to 100 bar

Hydraulic Pumps and Hydraulic Motors

Vane Pumps type V4

The basic design of model V4 is identical to the V3 fitted with variable displacement and pressure control.

The suction and delivery processes correspond to this.

The difference lies in the provision of dual vanes. As with the fixed delivery pump V2, we have 2 vanes in each rotor groove, resulting in two sealing edges and low contact pressure, due to hydraulic unloading in the transition range.

The second difference is ir. the type of control. The cam ring is fixed between two pistons 1 and 2 under system pressure, with an area ratio of approximately 1 : 2 (fig. 10).

A relatively weak spring 3 in the larger piston serves to protect the function when starting. It pushes the cam ring 4 at standstill and moves it into eccentric position at start-up.

The maximum operating pressure required is set at spring 5 of the pressure control valve 6. Spring 5 now holds the control spool 7 in the outlet position shown.

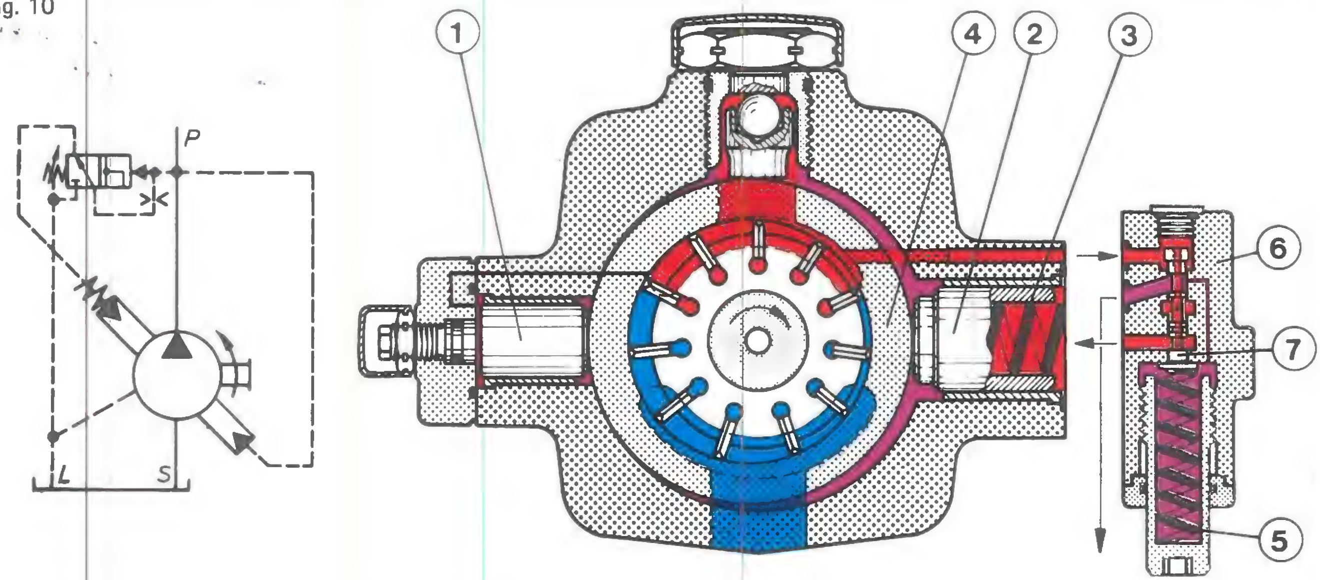
When the set pressure is reached, the spool in the control valve is pushed and thus the chamber behind the large piston 2 is connected to the tank via port L. The small piston 1 can now move the cam ring 4. The pump delivers only the quantity which the user requires.

As the cam is adjusted hydraulically and not by a spring, the Q-p-curve is practically vertical. It moves, remaining parallel, when higher pressures are set.

Different control devices can also be fitted, due to the hydraulic adjustment of the cam.

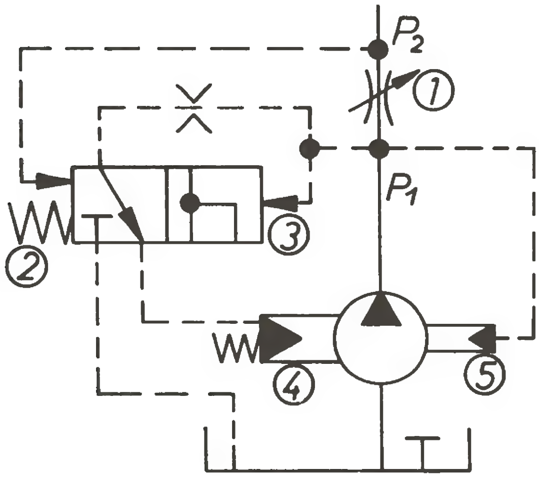
As an example of this, the flow controller with mechanical flow adjustment.

Fig. 10



Hydraulic Pumps and Hydraulic Motors

Symbol Flow Controller



The pressure differential ($p_1 - p_2$) of the throttle valve 1 connected in the P line works against the force of spring 2 on the spool of regulating valve 3.

The regulating spool works hereby as the pressure balance of a flow control valve (see also the chapter on flow control valves and flow regulating valves). It holds constant a pressure differential of 6 – 8 bar according to the spring force.

If, for example, the flow section at the throttle valve decreases, the pressure differential increases and pushes the regulating spool in the direction of the spring.

The control land is thus opened, causing the chamber behind the large adjustment spool 4 to be unloaded to tank and the cam ring pushed by the smaller adjustment spool 5 to smaller eccentricity. This carries on until the flow of the pump has decreased to the degree that control pressure differential of 6 ... 8 bar is reached again.

The following additional control units can be fitted:

– pressure controller with different types of control
– flow pressure controller
– horse power controller

Important technical details:

Displacement volume:	up to 125 cc/rev.
Operating pressure:	up to 160 bar

The most important characteristics of pumps type V3 and V4 are:

– Improvement of the energy balance by optimum and automatic adjustment of the flow to the actual requirement of the user(s).
– This generally causes reduction of the operating temperature with favourable effects, e.g. on the service life of the fluid and the seals.
– Use of a smaller oil tank possible.
– Simplification of the hydraulic circuit, as it is possible to do without pressure relief or shut-off valves.

Hydraulic Pumps and Hydraulic Motors

Radial Piston Pumps

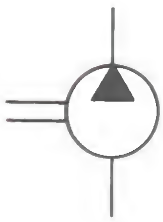
Radial piston pumps type R 2 with 3, 5 and 7 pump elements.

The pistons on a radial piston pump are arranged in star formation radially to the drive shaft. Movement of the working pistons is in a radial direction.

The pumps are valve or port controlled, with fixed or variable displacement. We must also differentiate between internal cam (pistons spring loaded inwards) and external cam (pistons spring loaded outwards).

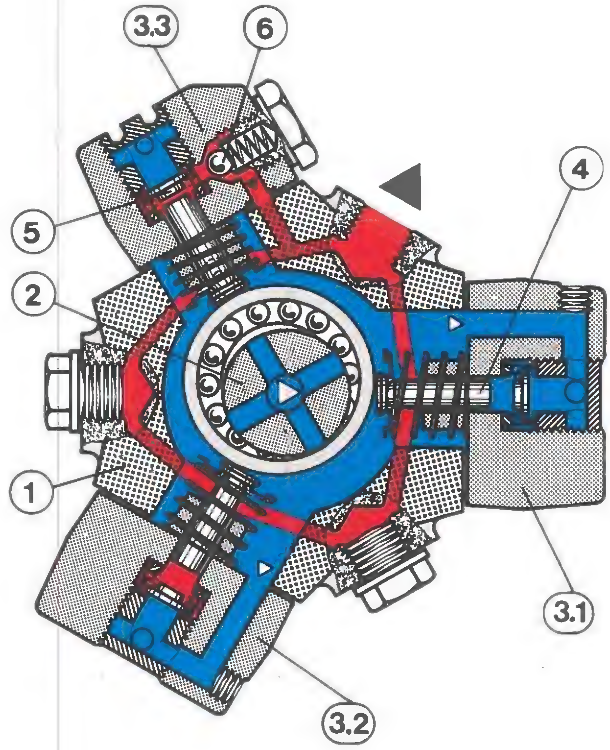
The pump shown in fig. 11 is valve operated, spring loaded inwards and self-priming; it has fixed displacement.

Symbol



It comprises mainly a housing 1, eccentric shaft 2 and the pump elements 3 with piston 4, suction valve 5 and pressure control valve 6.

Fig. 11



A pump element is a functionable one piston pump, screwed on to the housing. The pump elements are guides for the pistons, which are in turn pushed on

to the eccentric shaft by springs. Each piston carries out a double stroke per shaft rotation.

On rotation of the eccentric shaft, fluid (blue) is sucked through an axial bore in the shaft, fed externally through radial bores and then through lines to the suction valve.

The suction valve comprises a small valve plate, which is pushed by a weak spring externally on to a sealing land.

The piston chamber volume increases when the piston moves in the direction of the shaft centre. The resulting suction force causes the valve plate to rise from the sealing land and the piston chamber can fill with fluid (element 3.1).

If the piston is then pushed to the outside by the eccentric shaft, it pushes the valve plate on to the sealing land (element 3.2).

At the same time, the ball of pressure valve 6 rises from its seat (element 3.3).

The fluid can now flow from the individual pump elements to the pressure ports by means of channels in the housing.

The stroke volume is determined by the piston diameter and the number of pistons.

Since the power is dependent on both operating pressure and flow volume, maximum operating pressure also changes with the piston diameter used.

An uneven number of pistons is selected, so that irregularity of the flow volume is as low as possible.

Important technical data:

	Piston I.D. 8mm	10mm	12mm	14mm
Displacement volume in cc/rev. for 1 element	0.4	0.63	0.91	1.23
Operating pressure in bar up to	630	500	350	250

Fig. 12 also shows a valve operated, spring loaded inwards and self-priming radial piston pump type R4.

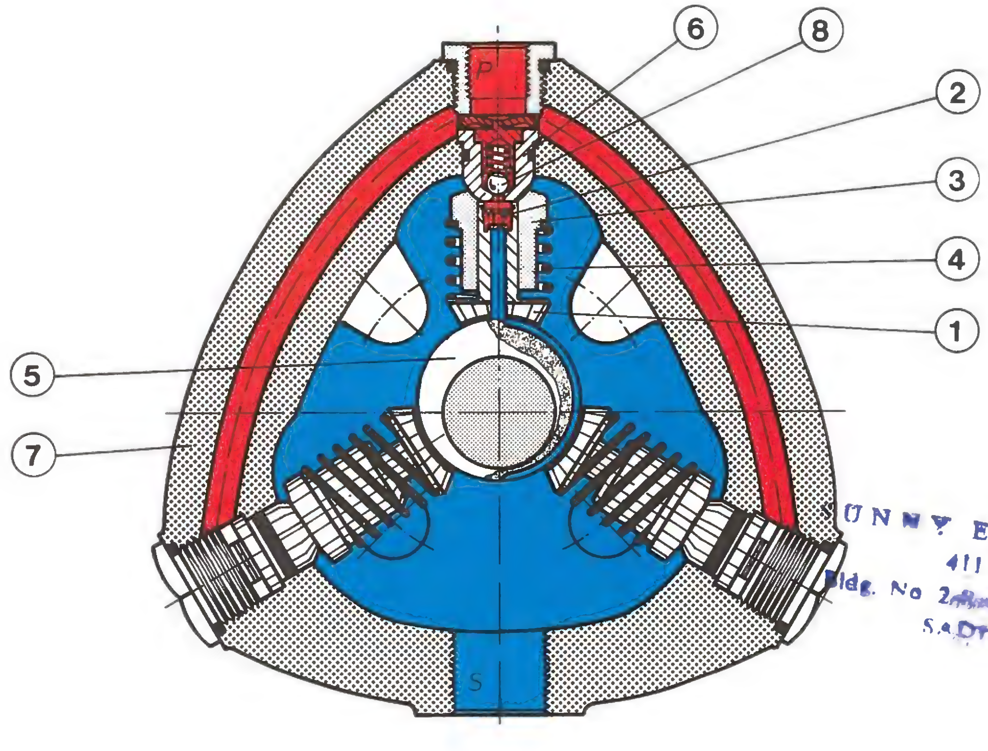
The difference from the pump described previously lies in the design of the piston elements. The hollow piston 1 with suction valve 2 runs in a cylinder 3 and is pushed on the eccentric 5 by means of a spring 4. The running surface of the piston corresponds to the eccentric radius.

The cylinder itself is ball shaped and pivots in the housing 7.

A pressure control valve 8 is fitted in this pivot. The actual piston element (cylinder, piston, suction valve) is therefore held freely by the spring between the eccentric shaft and the pivot (hydrostatically balanced piston bearing).

The piston chamber volume in the cylinder increases with downward movement of the piston.

Fig. 12



Hydraulic Pumps and Hydraulic Motors

Due to the suction, the valve plate lifts from the seal land. At the same time, the connection from the suction chamber to the piston is made by means of a radial groove in the eccentric.

The piston chamber is filled with oil by means of the groove and the hole in the piston.

When the piston moves upwards, the eccentric blocks the connection to the housing. The valve plate is pushed on to the seal land and the ball of the pressure valve is lifted from its seat. Fluid now flows on to the pump outlet.

The pump element carries out a pendulum movement during one revolution of the eccentric.

On this pump, 3, 5, or 10 pistons, also 3 different eccentrics are possible. For sizing purposes, the pressure elements can also be tapped individually.

Important technical data:

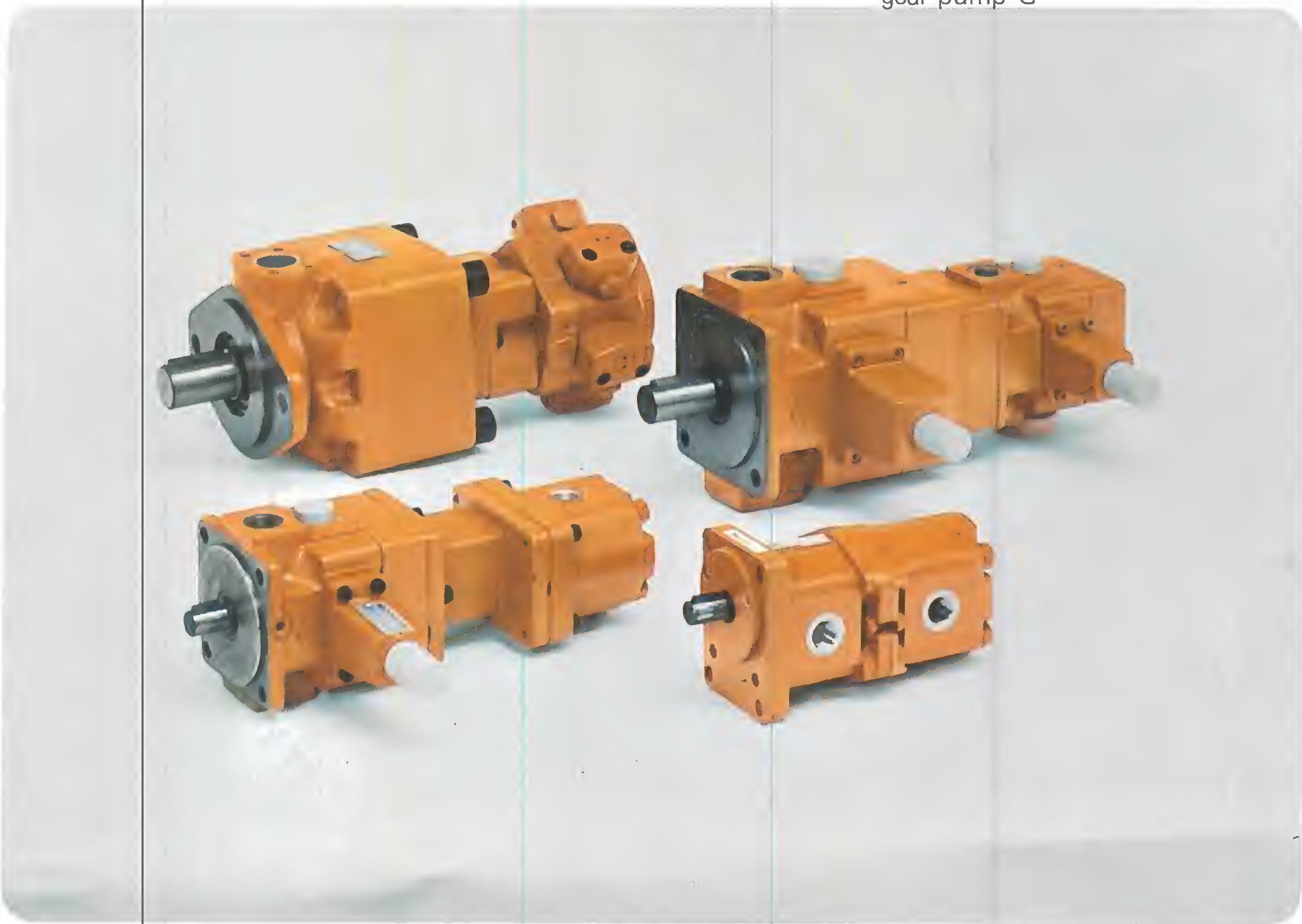
Piston I.D. (mm)	10		15
Displacement volume (cc/rev)	1.51	to	19.4
Operating pressure (bar)	700		500

Pump Combinations

The pumps which have been described previously can also be combined. The individual pumps are combined, using an intermediate flange. Only one drive motor is necessary for the pump combinations.

Where a pump combination is used, two separate circuits, for example, can be operated with one drive motor, or high pressure — low pressure circuit.

top left:	vane pump V2 + radial piston pump R2
top right:	vane pump V3 + vane pump V3
bottom left:	vane pump V3 + gear pump G
bottom right:	gear pump G + gear pump G



Hydraulic Pumps and Hydraulic Motors

Combinations:

Fixed Displacement + Fixed Displacement

Vane Pump V2 + Vane Pump V2

Operating pressure up to 175/175 bar
Flow up to (148 + 148) l/min

Vane Pump V2 + Radial Piston Pump R2

Operating pressure up to 175/630 bar
Flow up to (148 + 13) l/min

Gear Pump G + Gear Pump G

Operating pressure up to 250/250 bar
Flow up to (32 + 32) l/min

Variable Displacement + Variable Displacement

Vane Pump V4 + Vane Pump V4

Operating pressure up to 160/160 bar
Flow up to (200 + 200) l/min

Vane Pump V4 + Vane Pump V3

Operating pressure up to 160/100 bar
Flow up to (200 + 63) l/min

Vane Pump V3 + Vane Pump V3

Operating pressure up to 100/100 bar
Flow up to (63 + 63) l/min

Variable Displacement + Fixed Displacement

Vane Pump V4 + Radial Piston Pump R2

Operating pressure up to 160/630 bar
Flow up to (200 + 13) l/min

Vane pump V4 + Gear Pump G

Operating pressure up to 160/250 bar
Flow up to (200 + 32) l/min

Vane Pump V3 + Radial Piston Pump R2

Operating pressure up to 100/630 bar
Flow up to (63 + 13) l/min

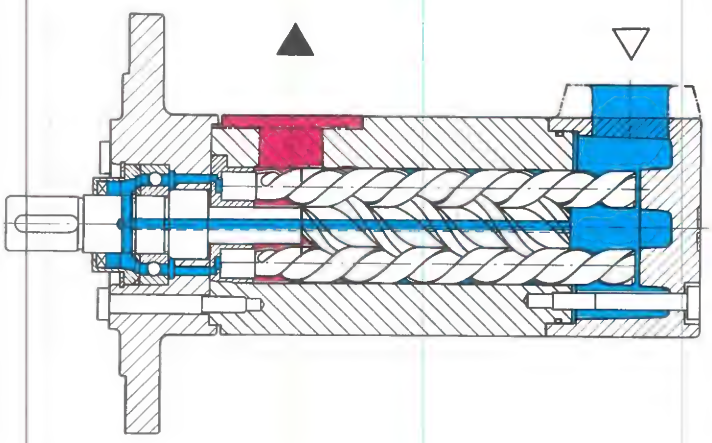
Vane Pump V3 + Gear Pump G

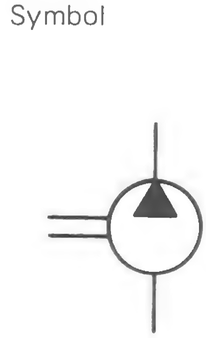
Operating pressure up to 100/250 bar
Flow up to (63 + 32) l/min

Hydraulic Pumps and Hydraulic Motors

Screw Spindle Pump (fig. 13)

Fig. 13





Two or more (here in fig. 13) spindles (worm gears) are fitted in a housing. The centre spindle with clockwise threads is driven by means of a shaft, and the rotary movement is transmitted to the two external spindles with anti-clockwise threads. Two threads of the external spindles, the housing and one thread of the drive spindle form an enclosed chamber. When the spindles are turned, this moves continuously from the suction side (blue) to the pressure side (red), without any change in volume. This gives a constant, uniform and smooth flow.

Axial Piston Pumps and Motors (Axial Piston Units)

Axial piston units are energy transformers, where the pistons are arranged axially in a cylinder drum.

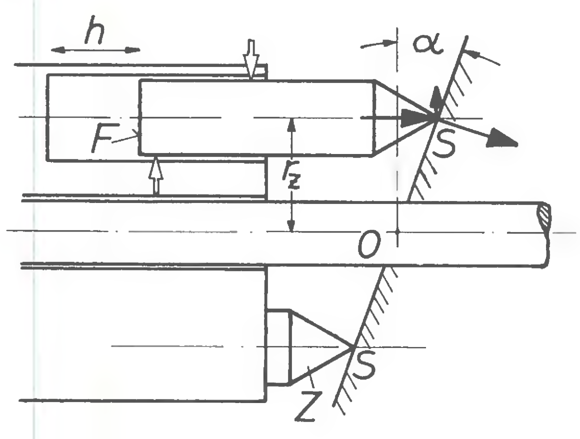
A distinction is made between the swashplate and bent axis designs.

The following diagrams show clearly the difference between the two models at the resolution of the piston forces at the transfer point and taking into consideration the nature of the torque.

On order to show this more clearly, the contact surfaces between piston and cam plate are shown with points.

At contact point S in fig. 14, the "hydraulic force" (pressure x piston area) is transformed into "mechanical force". The resultants of all annulus areas under pressure work vertically to the piston axis and move the piston to tilting position and create torque at the cylinder drum, which is fed to the drive shaft from the drum.

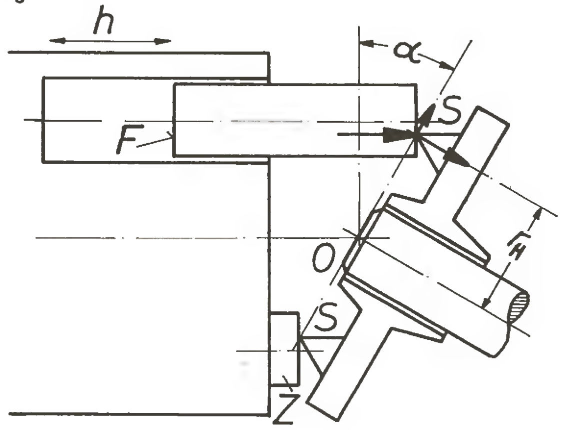
Fig. 14



Hydraulic Pumps and Hydraulic Motors

In fig. 15, the flat surfaces of the pistons rest on the cam plate fitted with points. The pressurised pistons and the cylinder drum remain without torque, which is created at the cam plate itself and is picked up directly there.

fig. 15



Both models are available with fixed and variable displacement.

The units described on the following pages are purpose controlled and can work in pump and motor operation without modification.

With **pump operation**, flow is proportional to the drive speed and the stroke volume.

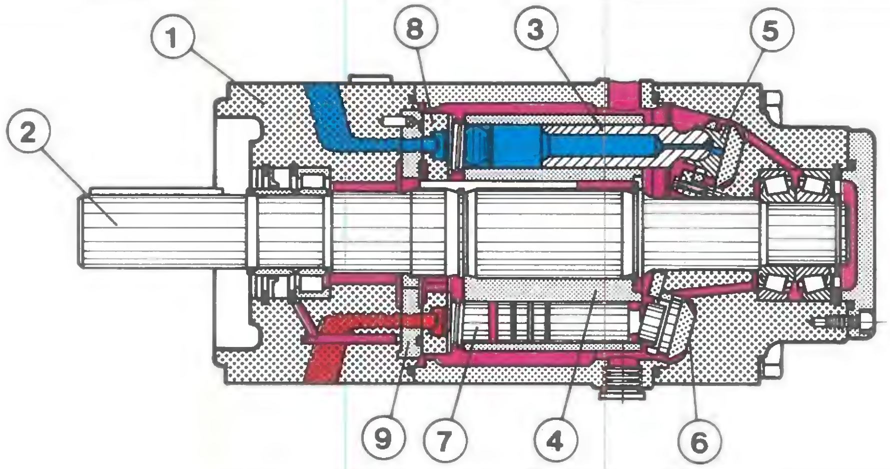
With **motor operation**, the speed of the driven side is proportional to the flow and inversely proportional to the stroke volume. The torque of the driven side increases with the pressure drop between the high pressure and low pressure sides.

Axial piston pumps/motors in swashplate design, fixed displacement (below) and variable displacement.

Hydraulic Pumps and Hydraulic Motors

Swashplate Design with Fixed Displacement

Fig. 16



Type A1F (fig. 16)

Nine pistons 3 are arranged in a circle, parallel to the drive shaft 2, in a fixed housing 1. They run in a cylinder drum 4, which is connected rigidly to the drive shaft by means of a key. The piston ends are designed as universal joints and fitted in slipper pads.

These are held on a 15° sloping surface 6 by thrust and retaining washers.

The sloping suface is part of the housing on fixed displacement units, and the slope is thus fixed.

When the drive shaft 2 is rotated (pump operation), the cylinder drum 4, collar bushes 7, cylinder cap 8, as well as piston 3 and slipper pad 5 also rotate. As the pistons are held on the sloping surface by means of the slipper pads, a piston stroke occurs in the cylinder drum when the drive shaft is turned.

The control and thus the supply and drain of the oil is by means of two kidney shaped grooves in the control plate 9, which is connected rigidly to the housing.

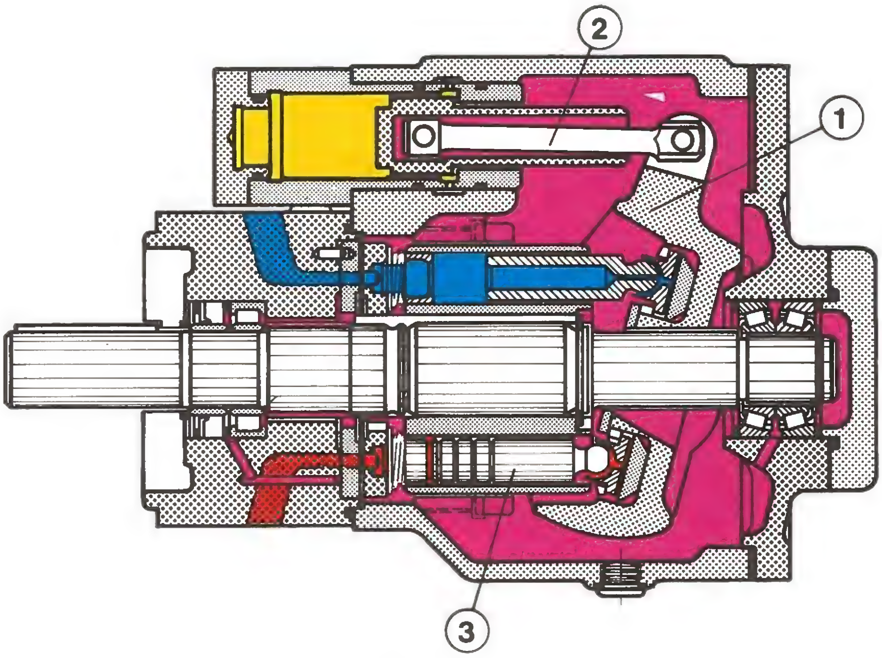
The pistons which move out of the cylinder drum are connected with the tank side (blue) by means of the control groove, and draw in fluid. The other pistons are connected to the pressure side (red) by means of the other control groove and push fluid by means of their stroke into the cylinder drum to the pressure port. One piston is always in the change-over range from suction to pressure side or vice versa.

Pressure fluid reaches the slipper bearing by means of a bore in the piston, and forms a pressure balance field there.

Hydraulic Pumps and Hydraulic Motors

Swashplate Design with Variable Displacement

Fig. 17



Type A1V (fig. 17)

On the model with variable stroke displacement, the sloping plane is to a certain extent a disc and no longer part of the housing.

The sloping plate 1 can move. It can be swivelled by means of an adjustment mechanism 2 through an angle of ± 15° to centre position. The pistons 3 carry out a certain stroke, related to the slope position, i.e. the swash angle. This stroke is the determining factor for the displacement volume. The piston stroke increases with increasing angle.

If the disc is in centre position (zero position) and thus vertical to the drive shaft, the piston stroke and thus the displacement volume are also zero.

Should the plate move above zero position while the speed remains unchanged, then the flow direction changes smoothly.

Design Characteristics of an Axial Unit in Swashplate Design (Type A1)

9 pistons/slipper pads
level control surface
through shaft
swash range from 0 up to ± 15°
short oil line.

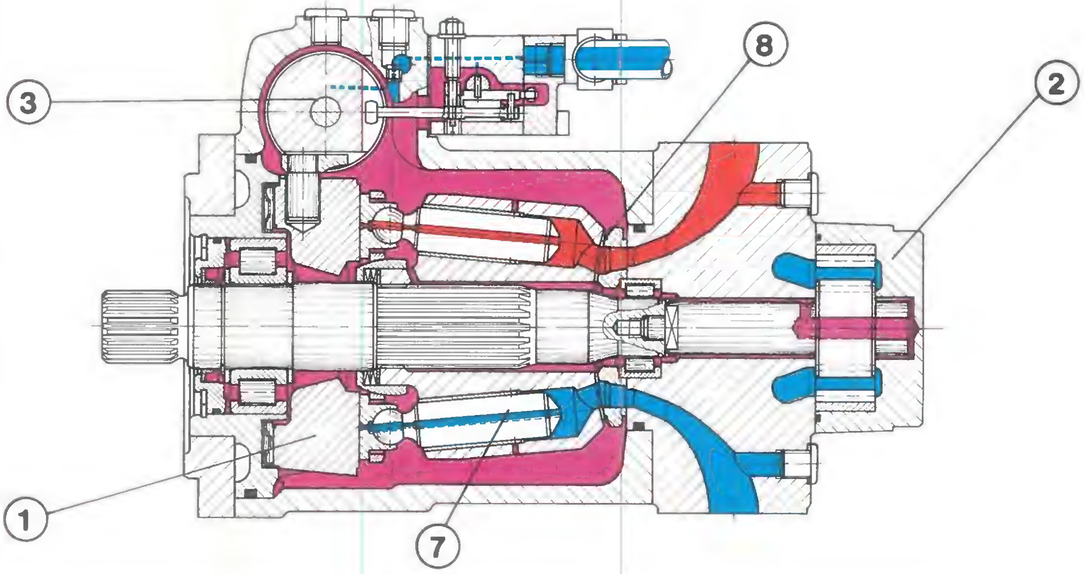
Special Characteristics:

- short swash times
- low weight
- good flow characteristics in zero position range
- compact
- second shaft extension possible, and thus additional pumps can be fitted
- well suited for reversing operation of large moments of inertia.

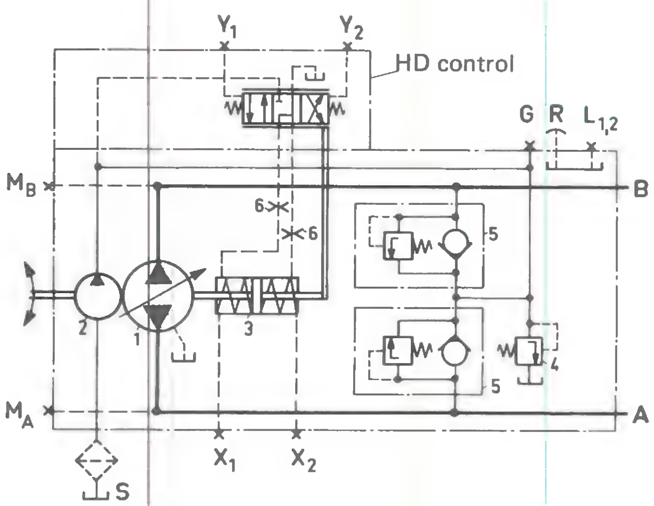
Hydraulic Pumps and Hydraulic Motors

Axial Piston Pump type A4V

Fig. 18



Symbol



Connection

A, B	working lines
G	pressure port for auxiliary circuits
L_1	leakage oil or oil filling
L_2	leakage oil or oil drain
M_A	measuring point working line A
M_B	measuring point working line B
R	bleed port
S	suction port for boost oil
X_1, X_2	ports for control pressures
Y_1, Y_2	remote control ports (e.g. hand pilot valve)

The axial piston variable displacement pump in swashplate design 1 is a primary unit ready for fitting; it incorporates auxiliary pump 2 for boost and pilot oil supply, adjustment unit 3, combined feed and pressure relief valves, and boost pressure relief valve.

One difference from the models already described is the tapered piston arrangement 7. With increasing speed, it increases the forces to retain the contact between the swashplate and the pistons. Also there is a spherical control surface 8 (advantage: self centering).

This variable displacement pump is designed for closed circuit and works in boosted operation, i.e. the fluid returning from the user is fed back to the pump under boost pressure. Leakage losses are replaced by the auxiliary pump.
Auxiliary pump 2 serves as a boost and control pump.

A boost pressure relief valve 4 is available to limit the maximum boost pressure.

Two built-in pressure relief valves 5 limit the pressure affecting the high pressure side and protect the unit from overloading.

The orifices 6 serve to set the stroke adjustment time

Hydraulic Pumps and Hydraulic Motors

Axial Piston Pumps/Motors in Bent Axis Design,
Fixed and Variable Displacement

Bent Axis Design with Fixed Displacement (fig. 19)

Drive shaft 2, cam plate 3, cylinder 4 with piston 5 and connecting rod 6, also control plate 7 are fitted in the fixed housing 1. The cam plate is vertical to the drive shaft. The cylinder with seven pistons and connecting rods is at an angle of 25^{0} to the shaft axis. The cam plate is linked to the cylinder by means of the piston rods. The cylinder rests on the central pin 8.

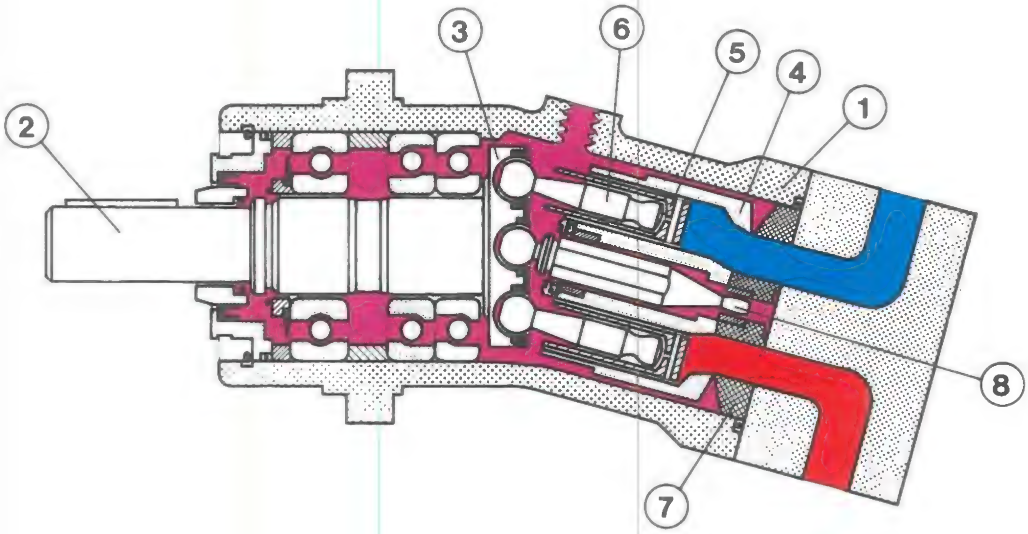
When the drive shaft 2 is rotated in pump operation, cylinder 4 also rotates by means of the connecting rods 6 and the pistons 5. As the pistons are held at the cam plate by means of the connecting rods, a piston stroke occurs in the cylinder drum when the drive shaft is rotated. The control plate has two kidney shaped grooves for supply (blue) and drain (red) of the pressure fluid.

In order to bring the cylinder drum on to the control surface of the control plate (also called port plate) without mechanical guides, this is designed spherically.

The drive to the pistons and cylinder is transmitted through the connecting rods, whereby the drag load (friction and inertia forces) do not impose side loading in the cylinders.

Type A2F

Fig. 19



Transverse forces at the cylinder drum are absorbed by the centre pin.

Bent Axis Principle with Variable Displacement
(fig. 20)

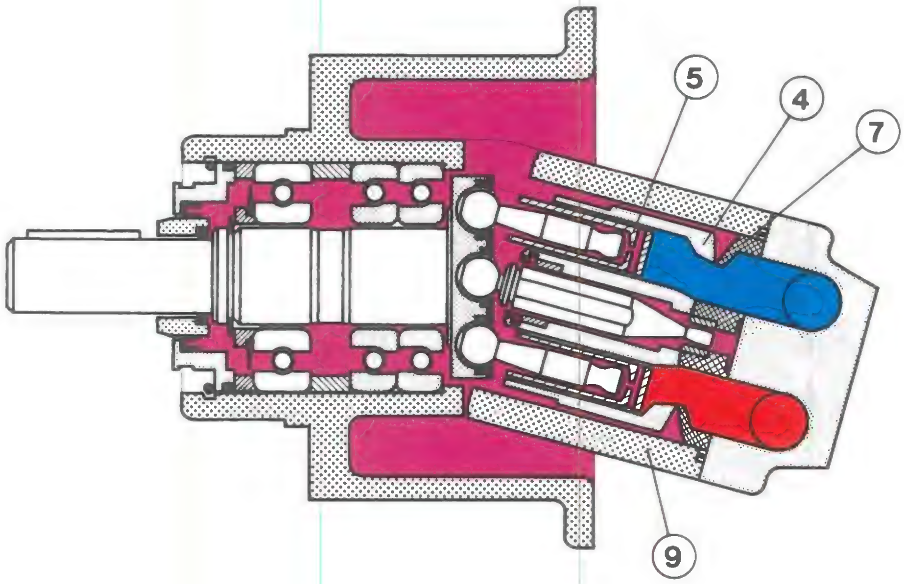
On the version with variable displacement, cylinder drums 4 with pistons 5, port plate 7 and housing are arranged, so that they can move.

The angle to the shaft axis can be altered between ± 25°.

The pistons carry out a certain stroke in the cylinder dependent on this tilt angle. The stroke and thus the displacement volume increases as the tilt angle increases. With the bent axis principle, the flow direction changes smoothly, when the body is swashed over zero position and the torque remains unchanged. If the tilt angle is zero, the displacement volume is zero.

The section shows an axial piston unit type A2V.

Fig. 20



Hydraulic Pumps and Hydraulic Motors

Design Characteristics of an Axial Piston Unit Type A2 in Bent Axis Design

7 pistons/connecting rods
spherical control surface and simple cylinder bearing
rigid shaft bearings
swash range from 0 to ± 25°
oil supply on the variable displacement unit by rotary joints

Special characteristics

- good suction even at high speeds
- favourable conditions in motor operation
- open construction possible
- good anti-cavitation characteristics

The operating of axial piston units, as mentioned in the preceding section, is described for pump operation.

With **motor operation**, the process is reversed. Fluid is fed to the axial piston unit. Transfer to mechanical torque is by means of operating pressure and stroke volume. In this case, pressure results from motor resistance (load resistance).

This motor resistance corresponds to the torque required at the driven side of the shaft. The driven speed changes according to the displacement. This is the oil quantity absorbed per time unit (generally in l/min) and corresponds to the flow of the pump.

Variable Displacement Motor, Type A6V, Bent Axis Design

This motor was designed specially for hydraulic drive with secondary control adjustment.

A complete adjustment device for a maximum swash angle range of 7° to 25° is fitted.

A lens shaped port plate 1 is fitted in place of the port plate shown in fig. 20. This is designed in such a way that it can be moved in a circular track.

Adjustment is by means of an adjustment piston 2 via a pin 3, which engages on the rear face of the port plate.

Control of adjustment piston 2 is by means of control piston 4, which is operated by pressurising or by a control solenoid, depending on the method of control selected. A separate pilot oil pump is not necessary, as the highest operating pressure at any time is drawn as adjustment oil from ports A or B (not shown in fig.).

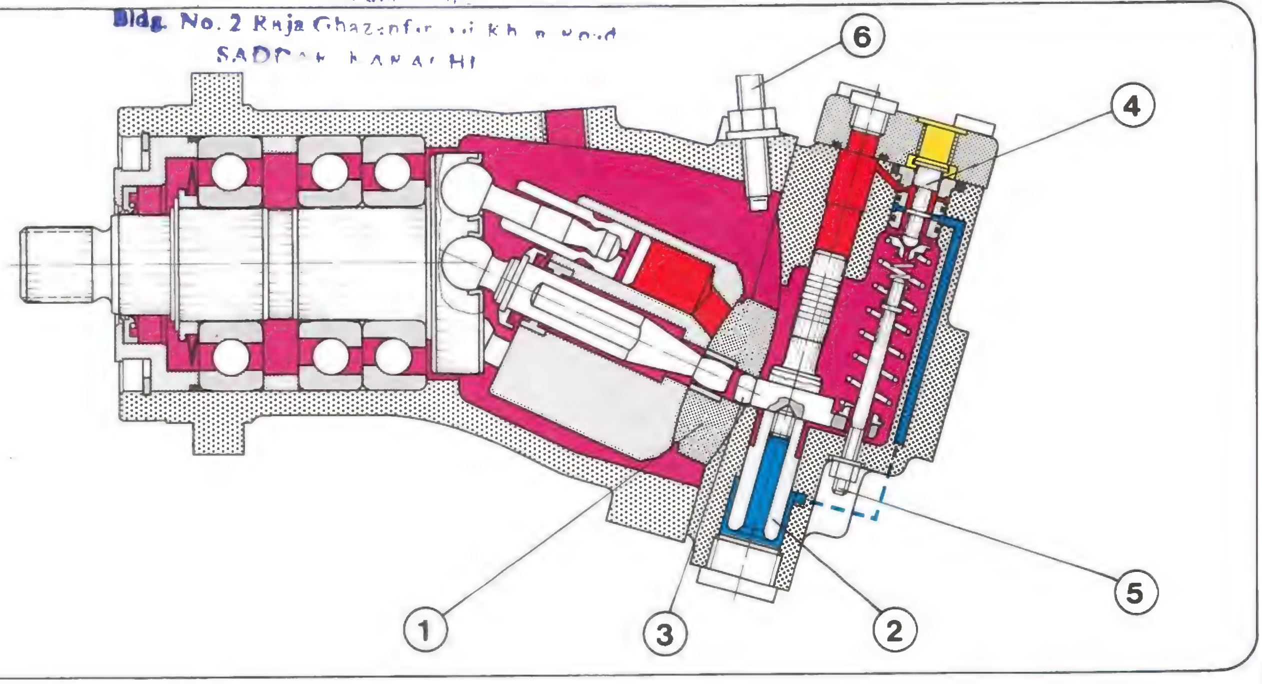
In order to guarantee that the adjustment functions perfectly, the high pressure and thus simultaneously the adjustment oil pressure must be at least 15 bar.

Variable displacement motor A6V with hydraulic control, pilot oil related (fig. 21).

5 adjustment screw for start of control

6 adjustment screw for limitation of tilt angle

Fig. 21



Hydraulic Pumps and Hydraulic Motors

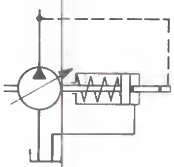
Control Devices

The control devices must also be mentioned in connection with variable displacement axial piston pumps and motors.

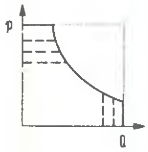
The following table shows the variety of control devices available.

LD HP Controller, direct operated

Symbol

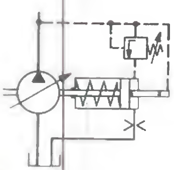


Curve

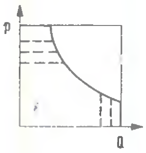


LDD HP Controller, direct operated with pressure cut-off

Symbol

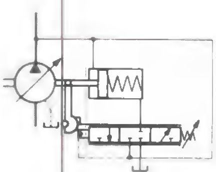


Curve

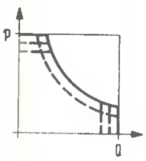


LV HP Controller, pilot operated

Symbol

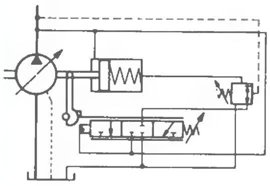


Curve



LVD HP controller, pilot operated with pressure cut-off

Symbol



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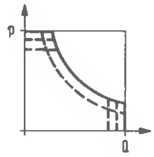
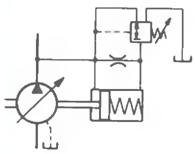


DRA Pressure Controller, built on directly

Symbol



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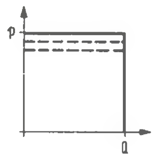
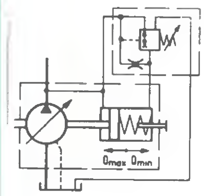


DRH Pressure Controller, hydraulically remote controlled

Symbol



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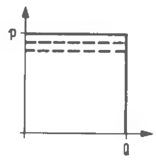
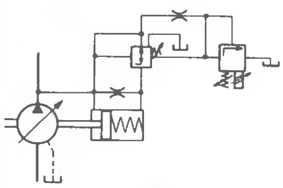


DRE Pressure Controller, electrically remote controlled

Symbol



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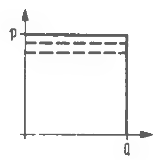
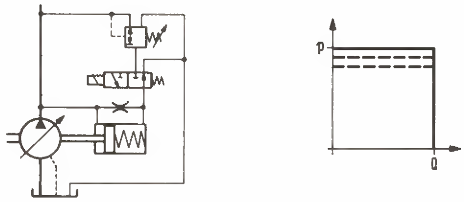


Hydraulic Pumps and Hydraulic Motors

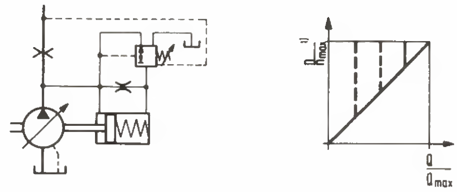
DRL Pressure Controller with unloading feature

Symbol | Curve



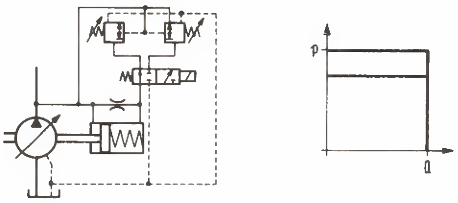
FÖ Flow Controller

Symbol | Curve



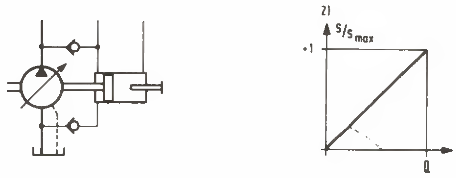
DRZ Pressure Controller with two switching points

Symbol | Curve



HM Hydraulic Control, flow related

Symbol | Curve

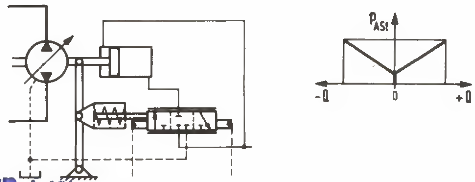


DRM Pressure Controller for variable displacement motor

Symbol | Curve

HD Hydraulic Control, pressure related

Symbol | Curve

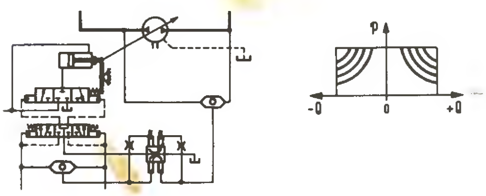


MA Manual Adjustment

Symbol | Curve

HDL Hydraulic Control, pressure related with HP limiter

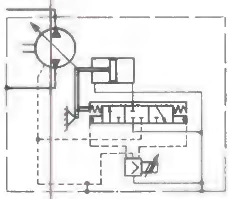
Symbol | Curve

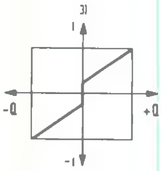


Hydraulic Pumps and Hydraulic Motors

HDS Hydraulic Control, pressure related with electro-hydraulic servo valve (1 off)

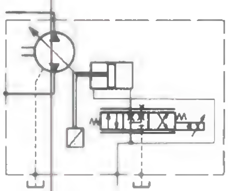
Symbol Curve

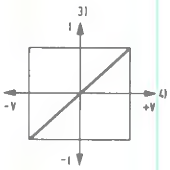




HSR Hydraulic Control, pressure related with electro-hydraulic servo adjustment and electrical feedback

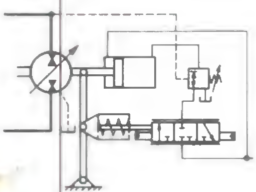
Symbol Curve

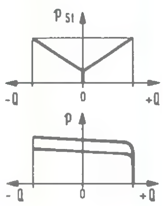




HDM Hydraulic Control, pressure related with overlapping mooring control

Symbol Curve

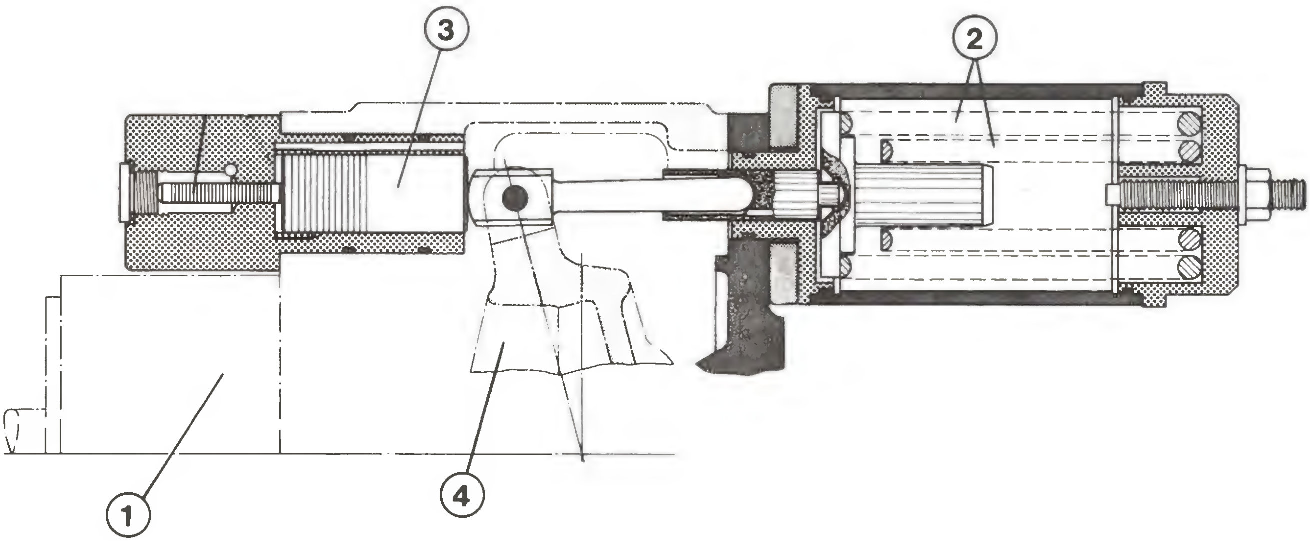




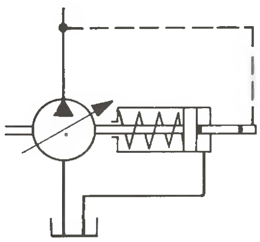
We now wish to look more closely at two of these controls, built on the axial piston unit in bent axis design:

HP Control, Direct Operated (LD) (fig. 22)

Fig. 22



Symbol



The direct operated HP control is flanged on to the rear of the axial piston unit 1. It comprises a spring assembly 2, against which a piston 3 affected by system pressure works.

The spring assembly presses the swashplate 4 in the direction of the increased flow.

With fixed drive speed, the HP control prevents the given drive power being exceeded by causing a decrease in flow with increase of operating pressure, so that the product of flow and pressure remains constant.

Hydraulic Control, Pressure Related (HD)

Adjustment of the stroke volume of the axial piston unit proportional to a pilot pressure is possible using this control.

The control comprises the pilot piston 1, control sleeve 2, measuring spring 3 and return lever 4, also a device 5 for setting the zero position (fig. 23).

Adjustment piston 6 is connected to the pump by means of a piston rod and the swashplate 7. The pilot piston is encased in the control bush and works against a pressure spring. This measuring spring forms the connection to the return lever, which pivots on the swashplate. The annulus area 8 of the adjustment piston is always affected by adjustment pressure. Pressurising of the piston area 9 is controlled by means of pilot piston 1.

If pilot pressure affects the right annulus area of the pilot spool, this is then pushed to the left against the force of the measuring spring, causing the adjustment piston cross area to be connected to the annulus area.

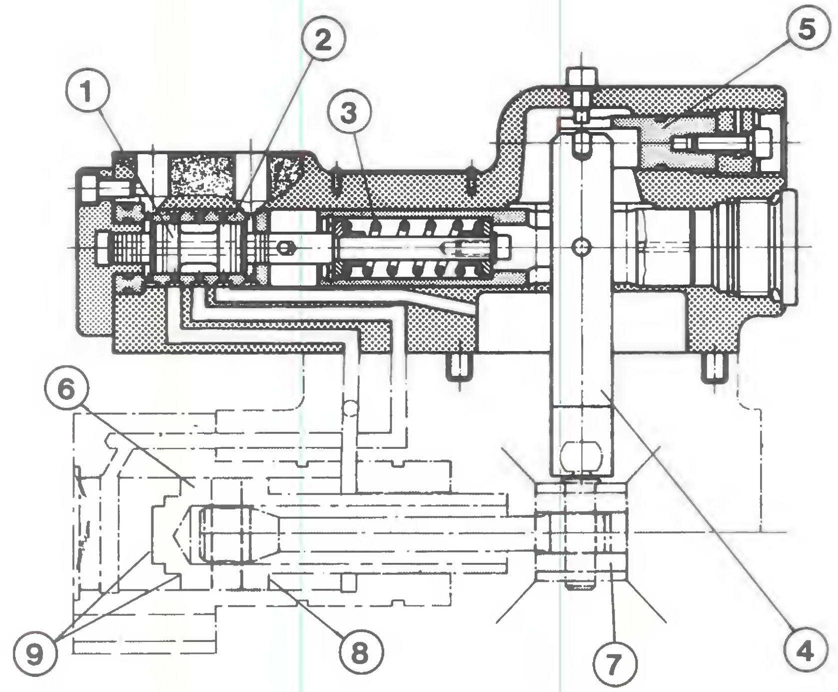
The adjustment spool moves to the right, until the force of the measuring spring corresponds to the hydraulic force at the pilot piston, and this is thus pushed into zero position once again.

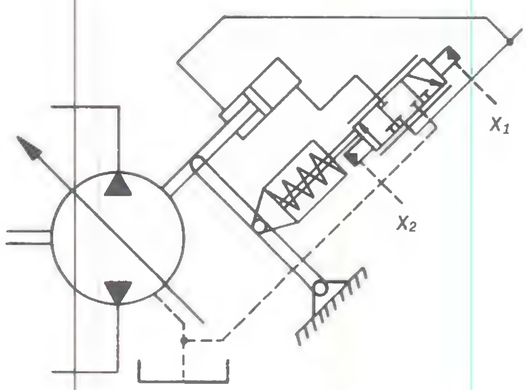
If pilot pressure affects the left annulus area of the pilot piston, then this is pushed to the right against the measuring spring and the gross piston area of the adjustment piston is connected to the outlet. The adjustment piston then moves to the left, until there is a balance between the hydraulic force at the pilot piston and the measuring spring force, and the pilot piston is in zero position once again.

The direction of flow of the pump reverses, as opposed to the process first described. A control pressure pump with 60 bar pressure is necessary for the control.

Hydraulic Pumps and Hydraulic Motors

Fig. 23





Important Technical Data for Axial Piston Units:

Operating pressure	continuous pressure 320 bar peak pressure 400 bar
Displacement volume	type A1 up to 250 cc/rev. type A2 up to 915 cc/rev. type E/C up to 2000 cc/rev.
Output torque	at Δp = 1 bar type A1 up to 4.05 Nm type A2 up to 14.54 Nm type E/C up to 31.832 Nm

Gear Motors

The design of gear motors type G corresponds exactly to the pumps with external gearing already described, the only difference being that the work method is reversed.

A torque is created at the gears by means of operating pressure and effective area.

Important Technical Details:

displacement volume:	up to 100 cc/rev.
operating pressure:	up to 250 bar
torque:	up to 18 Nm

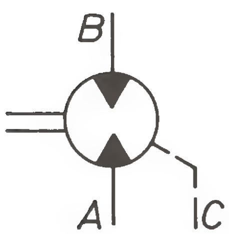
Hydraulic Pumps and Hydraulic Motors

Slow Speed Hydraulic Fixed Displacement Motors

The following models are shown above:

top left:	radial piston motor type RH 800, can be switched to half displacement
top right:	motor with built-on internal expanding brake or disc brake
below:	motor type S with rotating shaft

Symbol



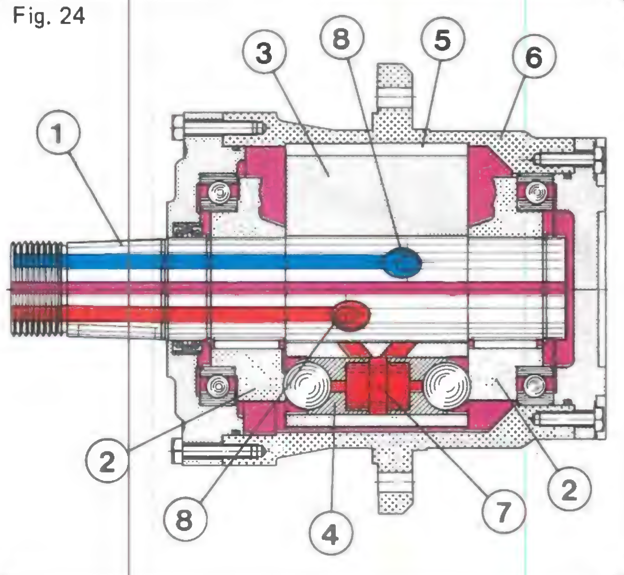
Slow speed hydraulic fixed displacement motors are designed as axial piston motors.

On the basic models, the motor is fixed to its rigid shaft, while the force is transmitted via the rotating housing. This version is used mainly for travel drives, winches, mixers, etc.

There is also a model with fixed housing and rotating shaft.

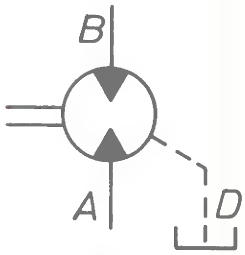
The basic principle of this motor design is to have an energy transformer, which, for example, drives the wheels of a vehicle **directly** or can be used as a cable winch.

Fig. 24



Axial Piston Motor Type CH 800 (fig. 24)

Symbol



The motor comprises a fixed shaft 1, two cam plates 2 fitted on both shaft ends, and the rotor/piston arrangement 3 (fig. 24).

The rotor between the two plates, with, according to motor type, 5, 8 or 9 bores fitted with 2 piston/ball combinations 4 lying opposite one another in each bore, is connected with the external housing 6 by means of key 5.

Rotor 3 and housing 6 can rotate round the fixed shaft 1. This movement is brought about by alternately pressurising the cylinder chambers 7.

The contra-rotating stroke movement of the piston/ball combination is changed to a rotary movement by its contact with the cam plates. Oil supply (red) and oil drain (blue) are by means of the fixed shaft in the motor.

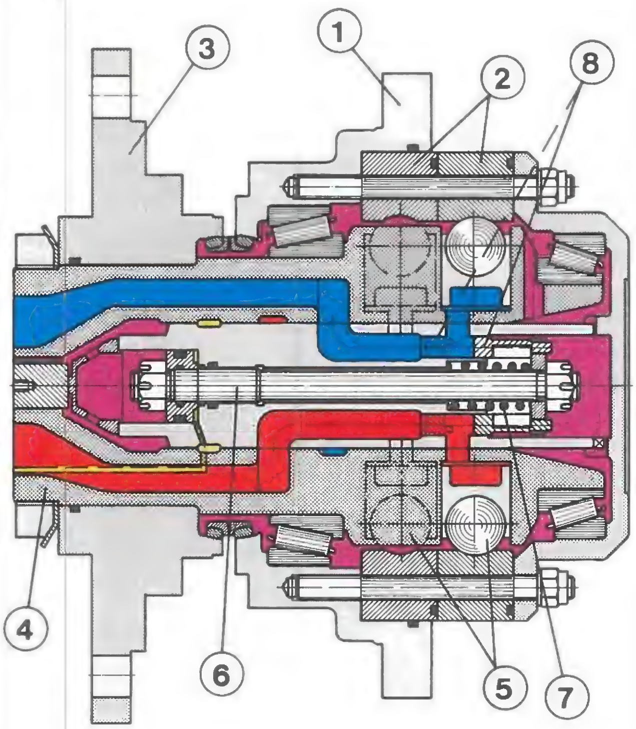
The control is by means of radial bores 8, arranged correspondingly in the shaft (pintel valve).

The motor can be reversed, even during operation, by changing direction of the oil supply and drain.

The piston/ball combination is radially arranged on this slow-speed motor.

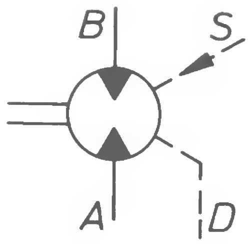
The motor comprises a housing with flange for power take-off, the radial cam plates 2 integrated in the housing, and the mounting flange 3 with shaft 4,

Fig. 25



Radial Piston Motor Type RH 800, can be switched to half displacement (fig. 25).

Symbol



in which 2 x 6 piston/ball combinations 5 and also the reversal unit 6 are fitted.

Supply and drain of the fluid is by means of the fixed shaft. Pressure affects the pistons and thus these are pushed with the ball on to the externally arranged track and cause the housing to rotate.

The twelve pistons are arranged in 2 rows of six. When pressure affects the annulus area (yellow) of the reversal unit 6, this is pushed to the left against the spring, by means of an external signal.

Hydraulic Pumps and Hydraulic Motors

Piston 8 thus blocks the flow of fluid to the right-hand row of pistons (shown dotted on the drawing) and unloads the pistons via the leakage port (purple).

Thus only the left-hand row of pistons is engaged, and the speed doubles while the oil supply remains constant. With the same pressure, only half the torque is available.

The hydraulic motor can be reversed by reversal of the oil supply and drain.

Free-wheeling is achieved, by unloading pressure from the pressure and return ports and putting pressure of 0.5 bar in the housing chamber. The connections between the piston/ball combinations are thus broken.

Important technical details:

Axial Piston Motor / Radial Piston Motor

geometric displacement	58...1000 cc/rev	817; 408.5 cc/rev
operating pressure	... 320 bar	... 350 bar
torque	... 4380 Nm	... 4150 Nm

Notes

Programme Summary

Hydraulic Cylinders (linear motors)

Cylinder type CD 70

Cylinder type CD 210

Cylinder type CD 250

Cylinder type CD 350

Cylinder type CD 70 / Cylinder type CD 210

Tie rod cylinder

- Spherical bearing at cylinder cap
- Rectangular flange at cylinder head
- Square flange at cylinder head
- Rectangular flange at cylinder cap
- Square flange at cylinder cap
- Central trunnion mounting
- Foot mounting with thrust key
- Foot mounting with o-ring seal for subplate connection
- Threaded fixing holes at cylinder head and cylinder cap
- Extended tie rod at cylinder head
- Extended tie rod at cylinder cap
- Clevis at cylinder cap

CD 70: up to 105 bar (related to piston dia.) piston dia. 25 – 200 mm

CD 210: up to 210 bar (related to piston dia.) piston dia. 40 up to 200 mm

Cylinder type CD 250 / Cylinder type CD 350

Screwed construction at both cylinder ends

Welded construction at cylinder cap and screwed construction at cylinder head

- Swinging eye at cylinder cap
- Swinging eye with sperical bearing at cylinder cap
- Flange mounting at cylinder head
- Flange mounting at cylinder cap
- Central trunnion mounting
- Foot mounting

CD 250: up to 250 bar piston dia. 40 up to 320 mm

CD 350: up to 350 bar piston dia. 63 up to 320 mm

Special cylinders

Rotary Actuators

Hydraulic Cylinders (linear motors)

Purpose and Models

Differential and synchronising cylinders with various mounting methods.

Hydraulic cylinders serve to carry out translatory (straight) movements and to transfer force by so doing.
The maximum cylinder force is dependent on the effective area and the maximum permissible operating pressure:

$$F = p \cdot A$$

It is constant from start to end of the stroke. The speed depends on the oil supply per time unit and the area. According to the design, a cylinder can apply compression or tension forces.

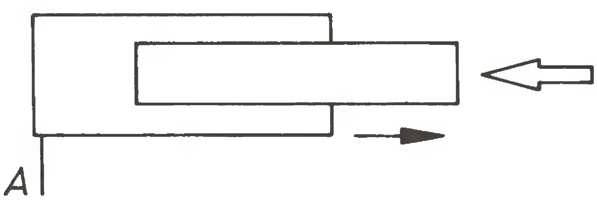
Summary of the types of hydraulic cylinders most frequently used:

Single Acting Cylinder

These cylinders can apply force in one direction only.

Plunger Piston or Plunger Cylinder

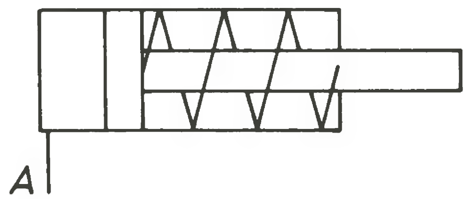
Symbol



When pressure affects the piston area via port A, the piston travels downwards (→). External force (⇐) is necessary for the piston to return.

Cylinder with Return Spring

Symbol



In this case, the cylinder also travels out hydraulically. Return stroke of the piston is achieved by means of the spring.

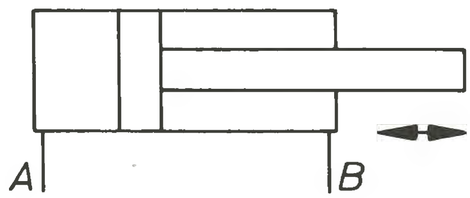
Double Acting Cylinder

The double acting cylinder can transmit force in both directions of movement.

Cylinder with Piston Rod on one Side

(differential cylinder)

Symbol



When oil is supplied via port A, the piston rod travels out, when supplied via port B, it returns.

Maximum forces are related to the effective areas:

travel outwards ⟶ piston area

travel inwards ⟶ annulus area

and the maximum operating pressure.

This means that forces are greater when travelling outwards than when travelling inwards.

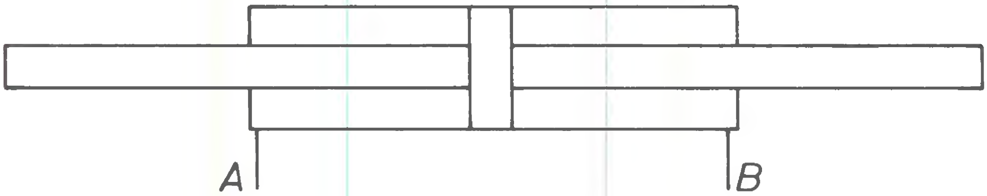
Hydraulic Cylinders (linear motors)

The volumes to be filled are identical in length, due to the stroke, and different according to the different areas.
Thus the speeds of movement have a reverse relationship to the areas:

this means: slow outward travel
faster inward travel

Cylinder with Piston Rods on Both Sides
(synchronous cylinder)

Symbol



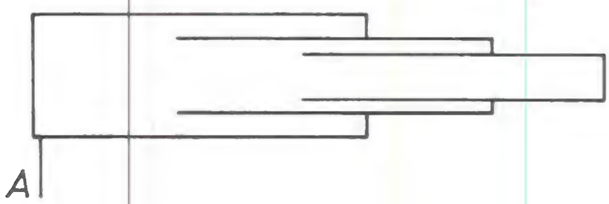
Where there is a through piston rod, the effective areas in both directions of movement are the same size. This results in equal values for the force, as for the speeds in both directions.

Telescopic Cylinder

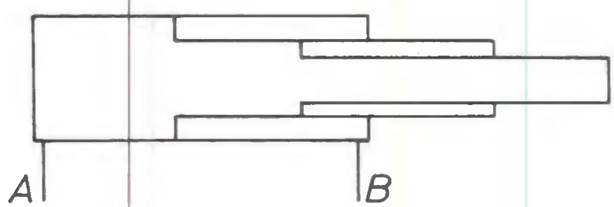
This is a special cylinder model with telescopic piston.

Symbol

single acting



double acting



A greater cylinder stroke with relatively small installation space can be achieved with this type of cylinder. The height is only a little more than one stage.

If pressure affects the pistons via port A, they travel outwards one after the other. The pressure varies according to the loading and the effective surface. Consequently, the largest stage travels outwards first.

At the same time, the pressure required increases with each stage, as the effective area decreases with the load remaining constant. When flow is constant, the speed of outward travel also increases from stage to stage.

When travelling inwards, the sequence is reversed, i.e. the small piston travels first.

The design used most frequently is the double acting cylinder with piston rod on one side, the differential cylinder.

We shall now look more closely at the standard series cylinder programme.

It can be sub-divided into 2 types:

1) tie rod design
2) screwed construction at both cylinder ends or welded construction at the cylinder cap and screwed construction at the cylinder head.

Hydraulic Cylinders (linear motors)

1) **Tie Rod Design, Cylinder** Type CD 70

The picture shows a tie rod cylinder with flange mounting at the cylinder cap.

The cylinder comprises mainly cylinder cap 1, cylinder tube 2, cylinder head 3, tie rod, piston 4 with piston rod 5, guide bush 6 and the mounting device (in this case, mounting flange 7). (fig. 1).

Cylinder cap, cylinder tube and cylinder head are assembled and then held together by 4 tie rods. Piston seal 10 is fitted between the piston side 8 and the rod side 9.

A "stick-slip" free wheeling motion (smooth) is achieved at the lowest speeds and low pressure, by the selected seal types and the surface quality of the cylinder tube, piston rod and piston rod guide.

This was taken into account according to the types of application for this tie rod cylinder, e.g. in machine tools.

An important dimension on differential cylinders is the

$$\text{area ratio } \varphi = \frac{\text{piston area}}{\text{piston annulus area}}$$

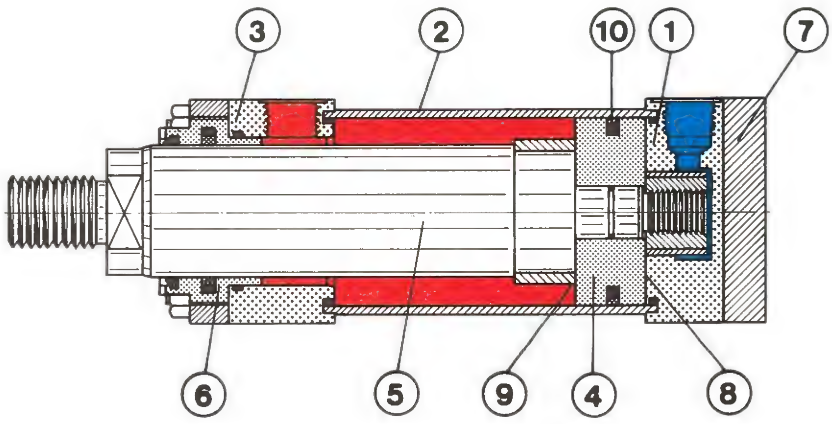
Piston annulus area = piston area − rod area

The maximum forces at inward and outward travel act according to this area ratio. The inward and outward travel speeds act inversely to the area ratio.

Important technical data:

operating pressure:	up to 105 bar (related to piston diameter)
piston diameter:	32 up to 200 mm
rod diameter:	18 up to 140 mm

Fig. 1



Hydraulic Cylinders (linear motors)

2) **Screwed construction at both cylinder ends**
or
Welded construction at cylinder cap and
Screwed construction at cylinder head

Hydraulic Cylinder CD 250/CD 350, Swinging Eye with Spherical Bearing at Cylinder Cap and Spherical Rod Eye at Cylinder Head

The section shows the model with welded cylinder cap and screwed cylinder head.
These cylinders are designed for operating pressures up to 250 bar (series CD 250) and up to 350 bar (CD 350) and are therefore robust (fig. 2).

The piston seal in this case is, as standard, a Chevron seal 1.

The rod guide is in the dry location 2 with rod diameter less than 100, and in the lubricated location 3 with rod diameter over 100.

Important Technical Data:

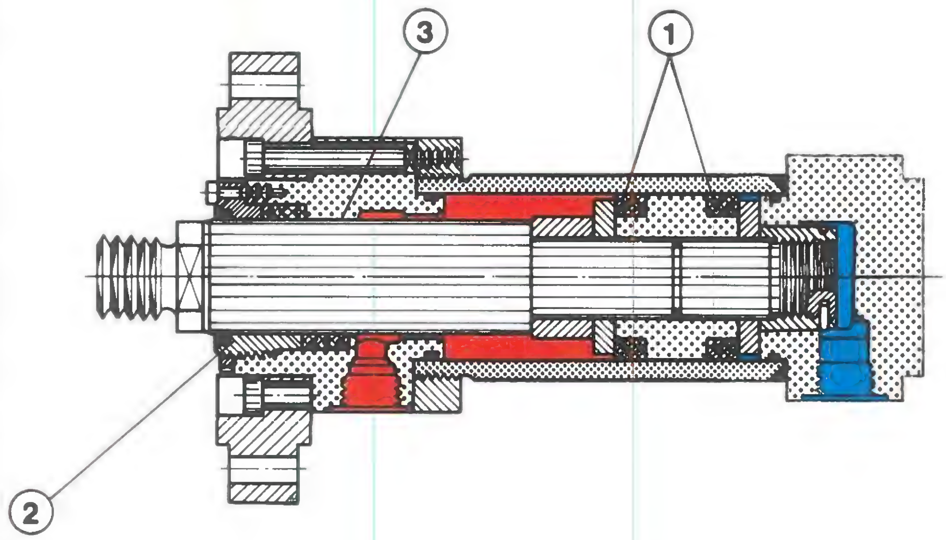
	CD 250	CD 350
operating pressure:	up to 250 bar	up to 350 bar
piston diameter:	40 – 320	63 – 320
rod diameter:	20 – 220	45 – 220

Alongside such series cylinders, a variety of special cylinders, designed for special applications, is also available.

The different installation and fixing possibilities are important for all cylinders, irrespective of the design.

They affect the stroke lengths in connection with piston rod diameter and load. Cylinders are usually designed for compression and tension forces. Reaction of cross forces and expansion should therefore be prevented.

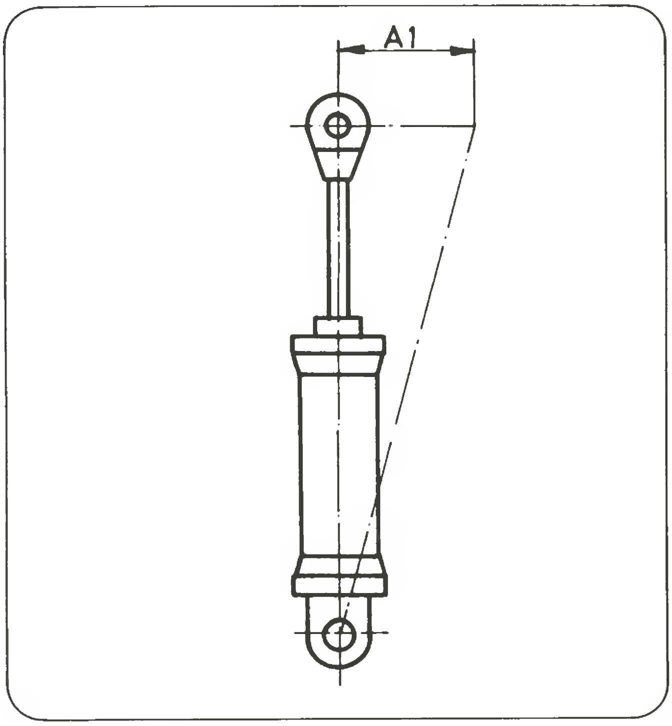
Fig. 2



Hydraulic Cylinders (linear motors)

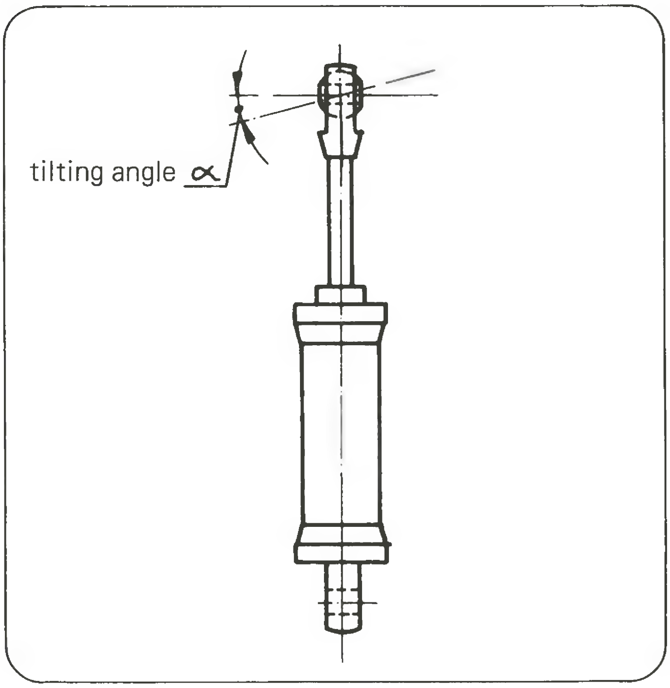
Fixing Methods:

1) **Swivel bearing at cylinder cap and rod eye with swivel bearing**



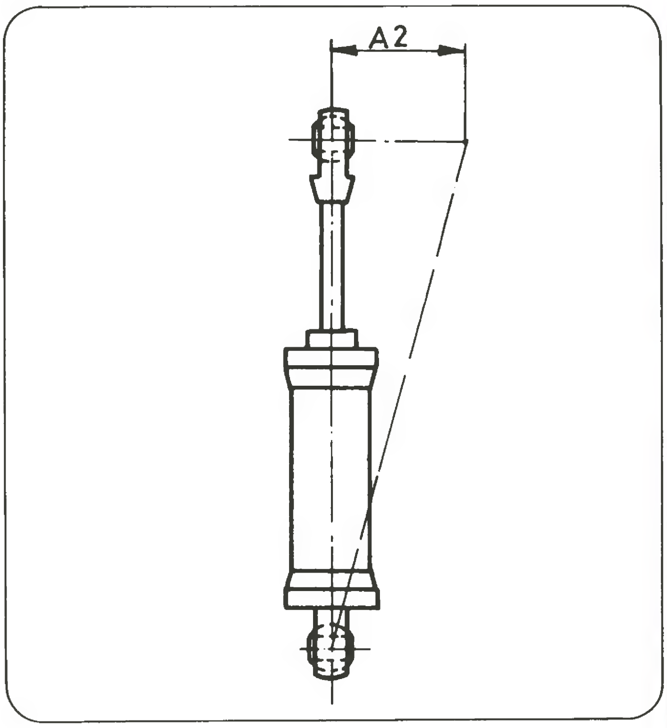
Axis movement possible in one direction only (in swash direction)

2) **Swivel bearing at cylinder cap and rod eye with spherical bearing**



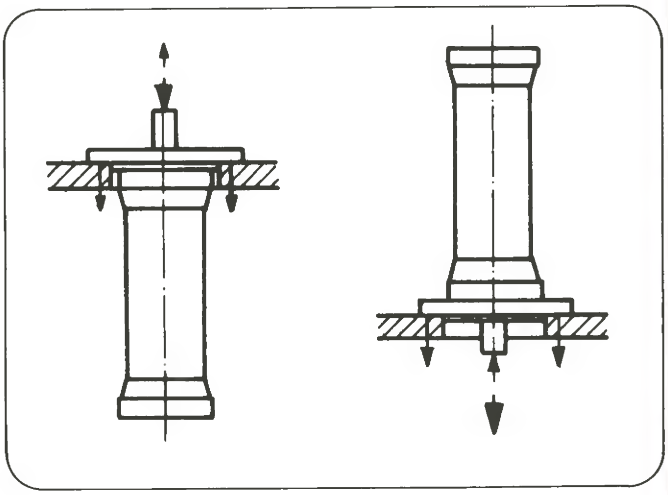
Inaccuracies in the alignment of the pivot bolts are also balanced out

3) **Spherical bearing at the cylinder cap and rod eye with spherical bearing**



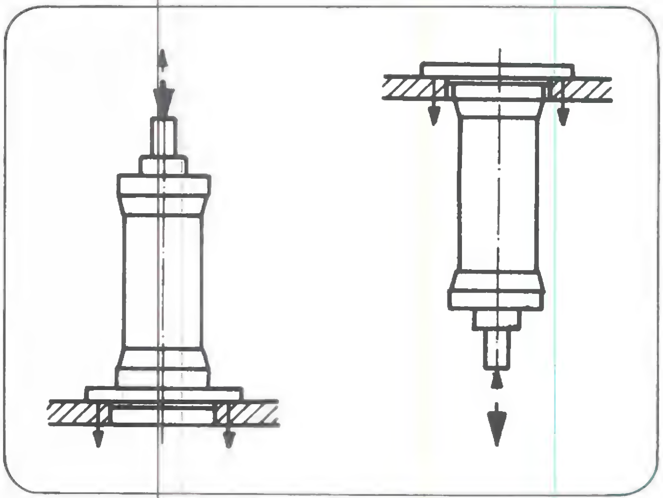
Lateral axis movement to actual swash direction

4) **Flange at cylinder head**



Hydraulic Cylinders (linear motors)

5) **Flange at Cylinder Cap**

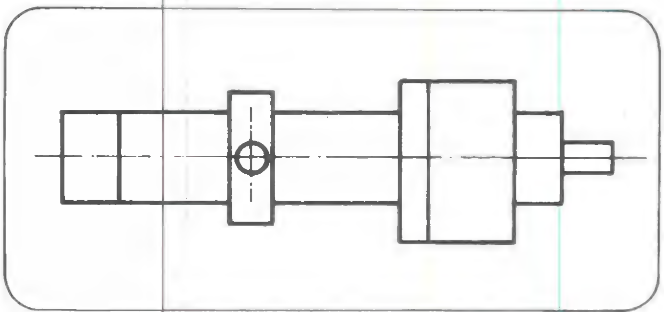


The following is valid for both installations:

a) mainly vertical installation

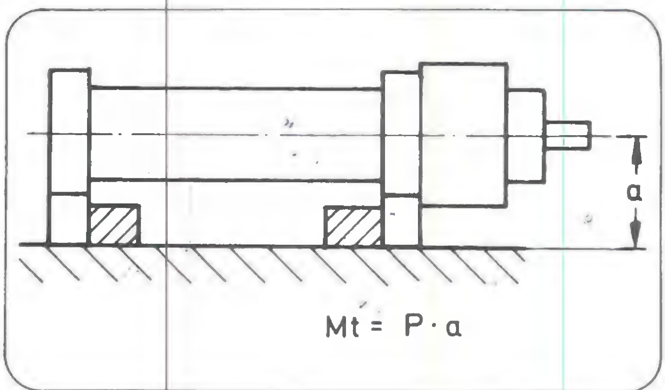
b) with main load (tensile or compression), the fixing screws at the flange must be unloaded

6) **Trunnion at Cylinder Tube**



For horizontal and vertical installation, trunnion position variable; trunnion in centre of tube = best centre of gravity

7) **Foot Mounting**



Torque occurs from the centre of the axis to the mounting surface. Pre-requisite: rigid design; good foot fixing to prevent loading of the fixing screws. Advantageous case: flexible bearing of cylinder and load, also load carried.

Buckling

Calculation of buckling is generally according to the "Euler" formula, since the piston rod is generally seen as the buckling member.

$$\textbf{Buckling load}\ K = \frac{\pi^2 \cdot E \cdot J}{s^2{}_k}$$

i.e. at this load, the rod buckles!

$$\textbf{maximum operating load}\ F = \frac{K}{S}$$

s_k = free buckling length (cm)

E = elasticity module (kp/cm^2)
= $2.1 \cdot 10^6$ for steel

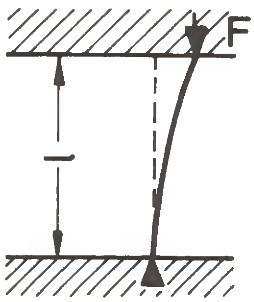
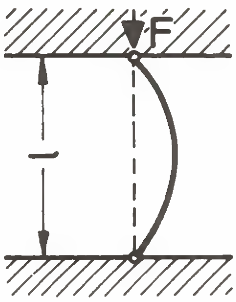
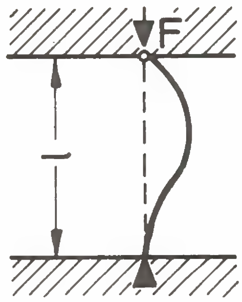
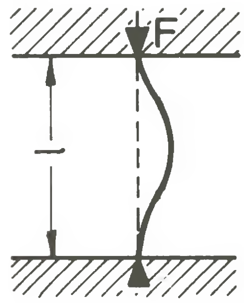
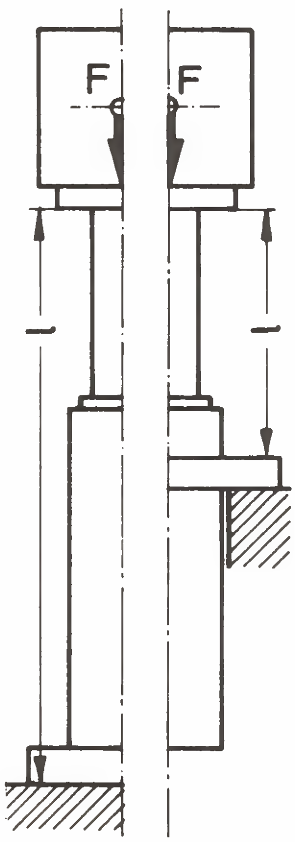
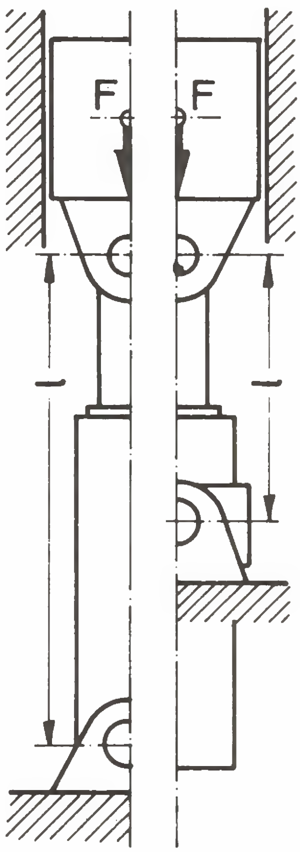
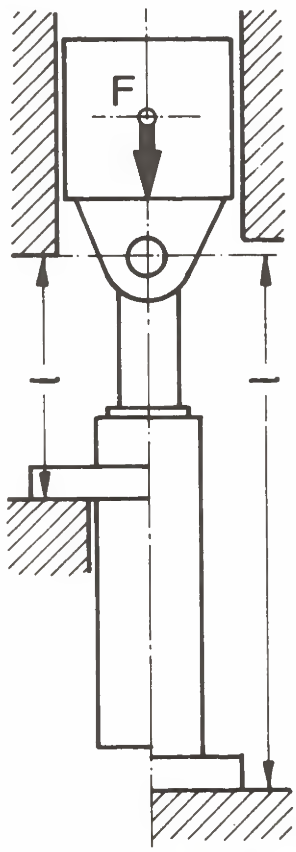
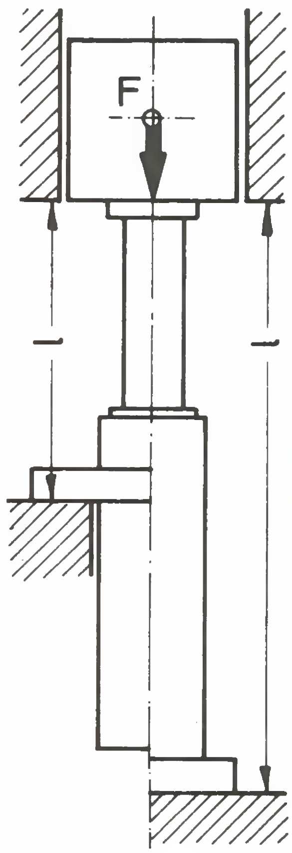
J = moment of inertia (cm^4)
$= \frac{d^4 \cdot \pi}{64} = 0.0491 \cdot d^4$
for circular section

S = safety (approx. 2.5 – 3.5)

The length to be stated as free buckling length must be taken from the Euler loading: see following table:

Reinforcement by the cylinder tube is not taken into account in the calculation. This gives an allowance for superimposed bending stresses due to the cylinder installation, the exact details being rarely known for standard cylinders.

Hydraulic Cylinders (linear motors)

	Case 1	Case 2 (basic case)	Case 3	Case 4
Euler loading	one end free one end rigidly fixed	two ends pivoted (and tigidly guided)	one end pivoted (and rigidly guided) one end rigidly fixed	two ends rigidly fixed (and rigidly guided)
Pictorial representation				
Free buckling length	$s_k = 2\,l$	$s_k = l$	$s_k = l \cdot 0{,}7$	$s_k = \frac{l}{2}$
Installation situation of hydraulic cylinder				
Notes			Guide load carefully, since side loading possible	

Hydraulic Cylinders (linear motors)

End Position Cushioning

Cushioning of the end position is necessary over a certain stroke speed. This refers to braking and deceleration of the stroke speed to standstill.

The kinetic energy resulting from this

$$E = \frac{m}{2} \cdot v^2$$

m = weight moved
v = stroke speed

must be absorbed by the end stop (cylinder head or cylinder cap).

Its capacity to absorb this energy depends on the elastic limit of the material.

It follows from this that a hydraulic brake function, i.e. end position cushioning, must be carried out where piston speeds exceed $v > 0.1$ m/sec.

The sectional diagram shows an adjustable end position cushioning on the piston side (fig. 3).

Cylinder cap 1, cylinder tube 2,
Piston 3

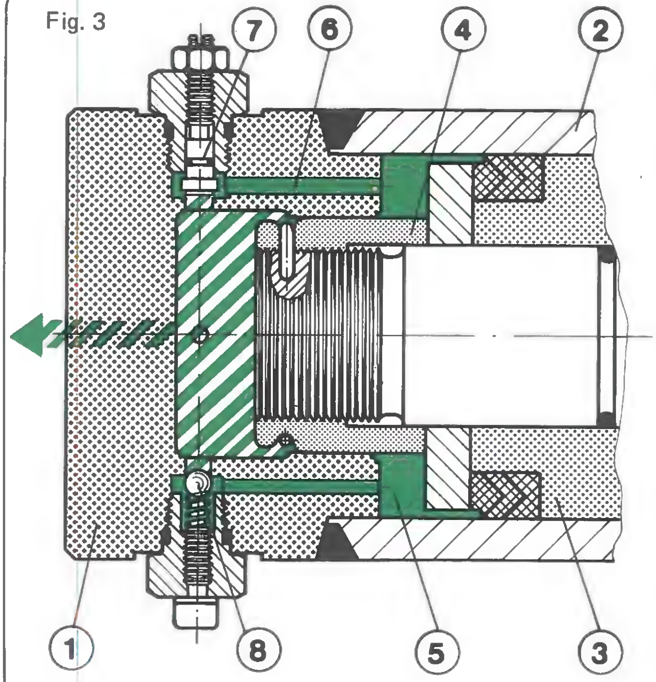
The cylinder piston 3 is fitted with a tapered end position cushioning.

If the piston with cushioning bush travels into the bore in the cylinder cap, the section for the fluid flowing out of the piston chamber 5 decreases, until it finally closes completely. The fluid must now drain from the piston chamber by means of the bore 6 and the adjustable throttle valve 7.

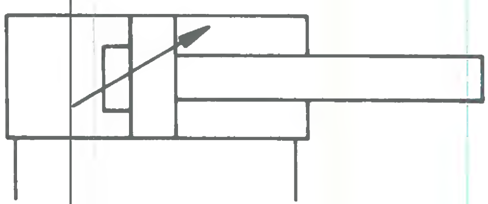
The cushioning effect can be regulated at the throttle valve. A smaller section at the throttle valve results in stronger cushioning.

Check valve 8 is fitted to assist travel from the end position. The throttle position is thus by-passed at outward travel.

Fig. 3



Symbol



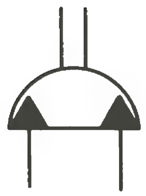
Cylinder with adjustable cushioning on the piston side.

Actuator

Actuator (swash motor), **applications**

Swash Motor Type B with Hollow Shaft

Symbol



Swash motors type B serve to create torques by means of a limited swash angle.

The swash motor comprises a housing 1, a double acting spool with geared rod 2 and a drive pinion 3 (fig 1).

The middle part of the piston, designed as a geared rod, is engaged with the drive pinion.

When one piston side is affected by pressure, the piston moves, causing the drive pinion to move with it.

The torque at the drive shaft depends on the operating pressure, and the speed of rotation on the fluid supplied.

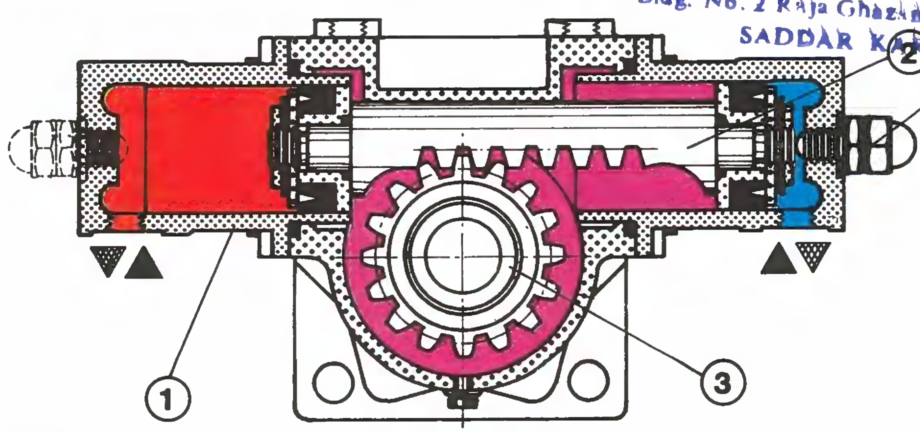
The piston stroke and thus the swash angle can be limited by means of adjustment screw 4.

The actuator transmits a constant torque over the complete range. There are other designs, such as parallel pistons, rotary vanes or, for example, crank gears, where the size of the maximum torque depends on the angle of actuation.

Important technical details:

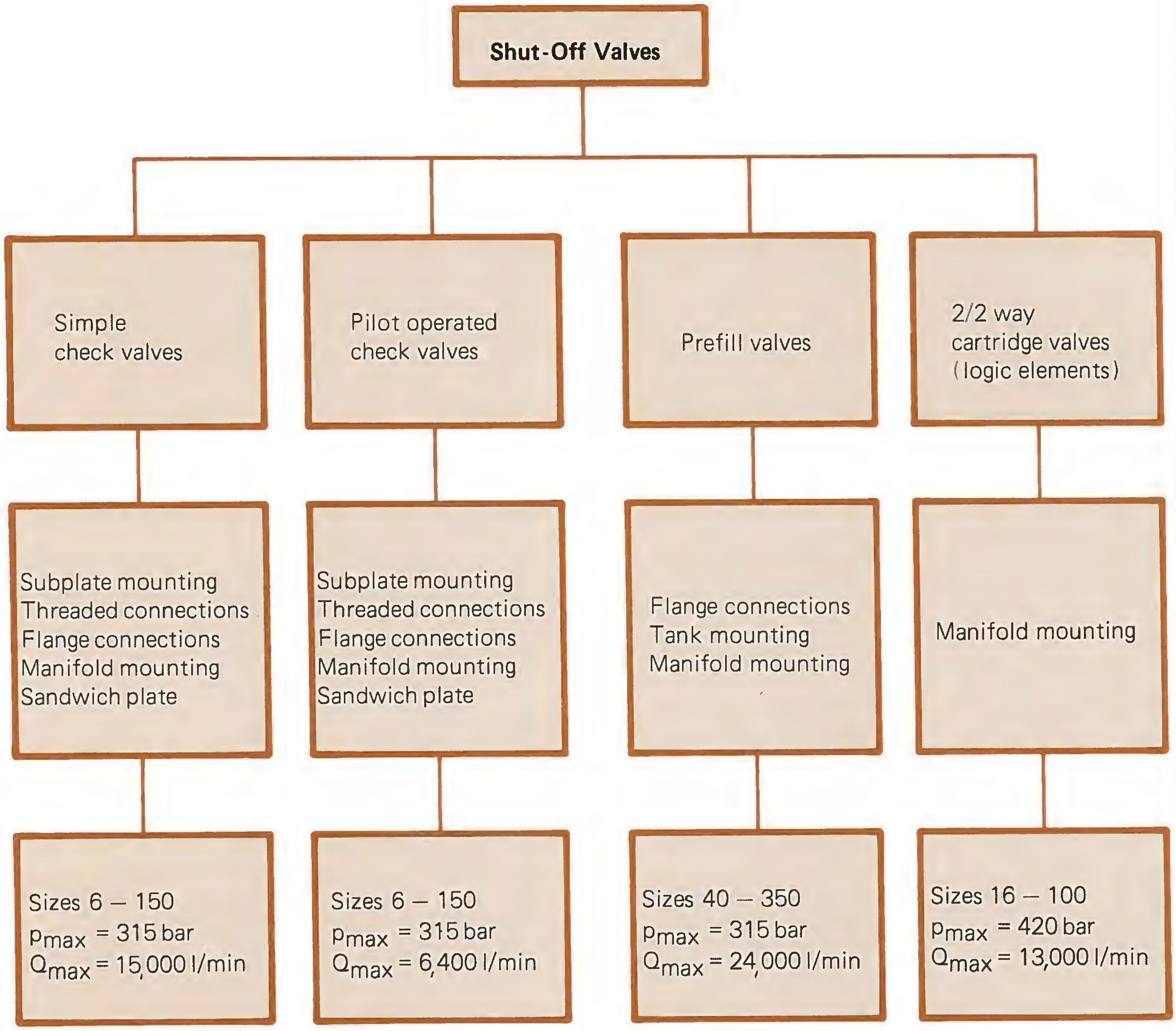
swash angle	up to 180 °
torque	up to 2650 daNm
operating pressure	up to 160 bar

Fig. 1



Notes

Programme Summary



Shut-Off Valves

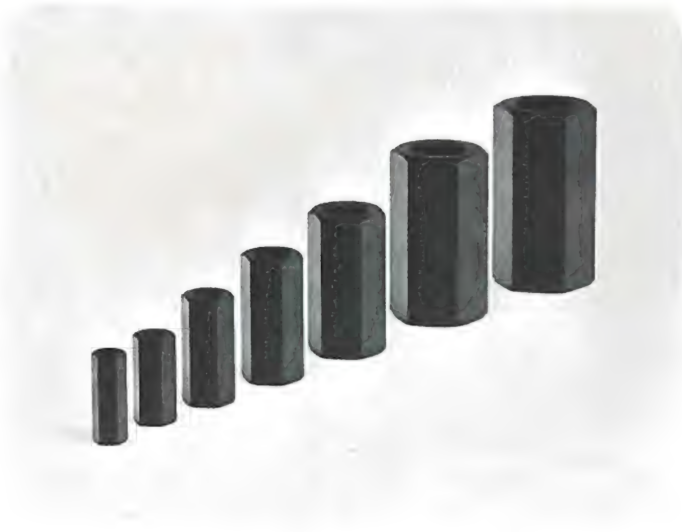
Purpose

In a hydraulic system, shut-off valves serve to check flow in a preferred direction and allow free flow in the opposite direction. They are designated as check valves.

The shut-off valves are designed as poppet valves and therefore provide leakfree closure.

A ball or poppet is generally used as the closing element.

Simple Check Valves



Check valve of different sizes for direct line mounting

Symbol

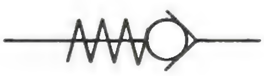
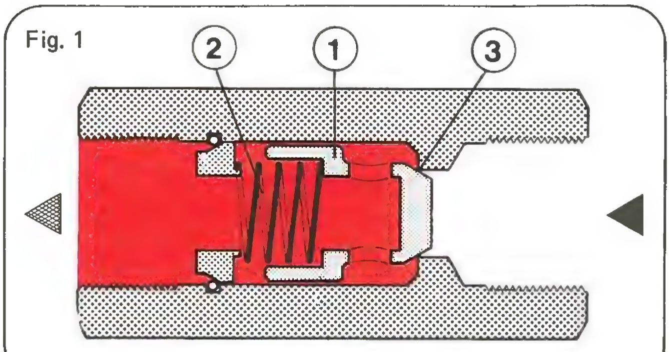


Fig. 1



The sectional diagram shows a simple check valve (as in the photo), where the closing element is a poppet 1, which is pushed on to the seat 3 in the housing by means of the spring 2. The mounting position for this valve is optional, since the spring always holds the closing element on the seat (fig. 1).

When there is flow through the valve in the direction of the arrow, the poppet is lifted from its seat by the fluid pressure and allows free flow. In the opposite direction, the spring and the fluid push the poppet onto the seat and close the connection.

The cracking pressure depends on the spring selected, its compression and the pressurised poppet surface area, and is generally between 0.5 and 3 bar, depending on the application.

Valves with a low cracking pressure are used at present to by-pass a throttle position in one direction.

When used as a by-pass valve to by-pass a return line filter, cracking pressure of, for example, 3 bar is found suitable to limit the pressure reached due to contamination.

A check valve without spring must always be fitted vertically. The closing element then remains on the seat in the no flow condition due to gravity.

It should be mentioned that the closing element can also be a disc or, for example, a hollow poppet.

Important Technical Data

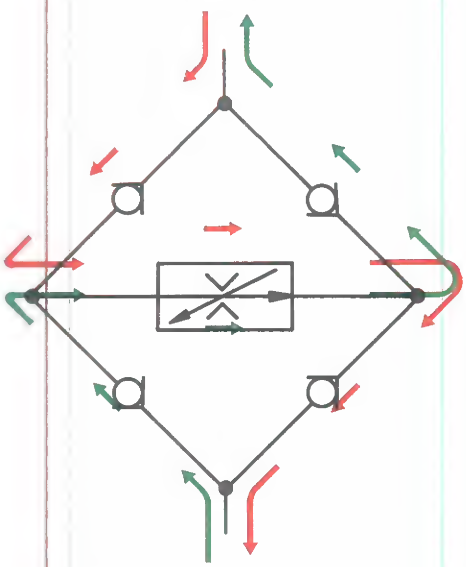
Size	6 to 150
Flow	up to 15,000 l/min (at v_{oil} = 6 m/sec.)
Operating pressure	up to 315 bar
Cracking pressure	without spring; 0.5; 1.5 or 3 bar

The so-called "rectifier circuit" is achieved by corresponding arrangement and connection of 4 check valves. It is used mainly in connection with flow control valves. With this circuit, the fluid must flow through the valve in the same direction at forward flow (red) and return flow (green) (figs. 2 and 3).

Shut-Off Valves

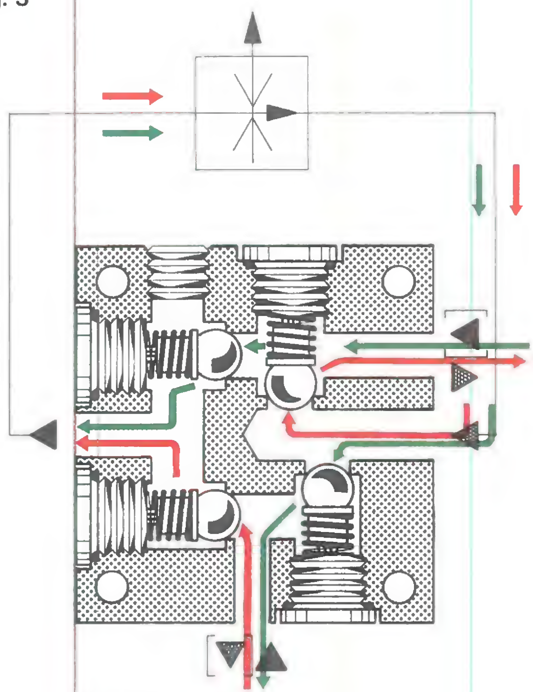
Circuit with Flow Control Valve and Details of Flow Direction

Fig. 2



Sectional Diagram of the Adaptor Plate type Z4S with details of flow direction.

Fig. 3



Shut-Off Valves

Pilot operated check valves

Left: *pilot operated check valve with threaded connections*

Right: *double throttle/check valve, sandwich plate design*

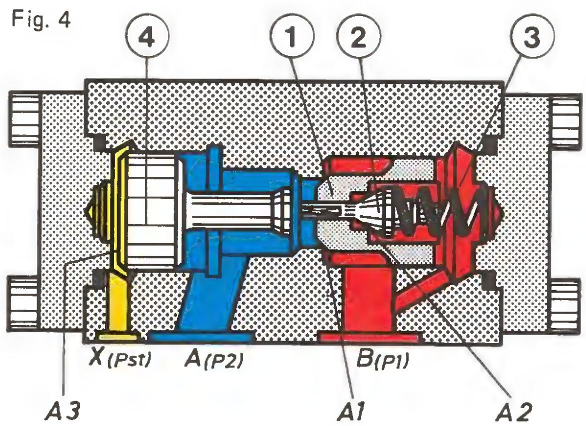
As opposed to the single check valves, pilot operated check valves can also be opened in the direction of closure.

The valves are used, for example:

- to seal working circuits under pressure
- as a protection against the load dropping, if a line should break
- against creep movements of hydraulically stressed users.

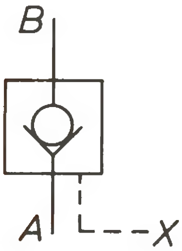
Model without leakage port

Fig. 4



The sectional diagram shows valve type SV, without drain port, with opening poppet.

Symbol for valve type SV



There is free flow in direction A to B, from B to A the main poppet 1 with pilot poppet 2 is held on its seat by system pressure in addition to the spring 3 (fig. 4).
When pressure affects control port X, the pilot spool 4 is pushed to the right. Firstly the pilot poppet 2 and then the main poppet 1 are thus pushed from their seats. Oil can now flow through the valve from B to A also.
The pilot spool causes a cushioned decompression release of the fluid under pressure. There are therefore no switching oscillations.

A certain minimum pilot pressure is required, so that the valve can also be safely controlled by means of the control spool.

Required pilot pressure at port X:

$$p_{St} = p_1 \cdot \frac{A_1}{A_3} + C$$

Pressure at port B:

$$p_1 = \left(p \cdot \frac{A_K}{A_R} + \frac{F}{A_R}\right)$$

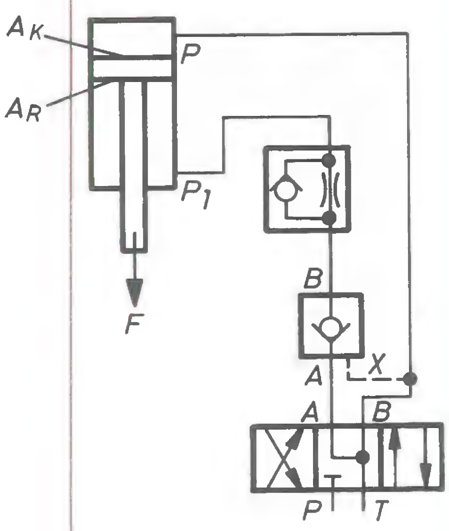
(see the following pages for explanation of the letters)

Shut-Off Valves

A_1 = main poppet surface area (cm²)
A_3 = control spool surface area (cm²)
C = coefficient for spring and friction (bar)
A_K = piston surface area at cylinder (cm²)
A_R = annulus area at cylinder (cm²)
F = load at cylinder (daN)
A_2 = surface area of pilot unloading poppet

The following circuit diagram shows once again the conditions given for control pressure in the equation (fig. 5).

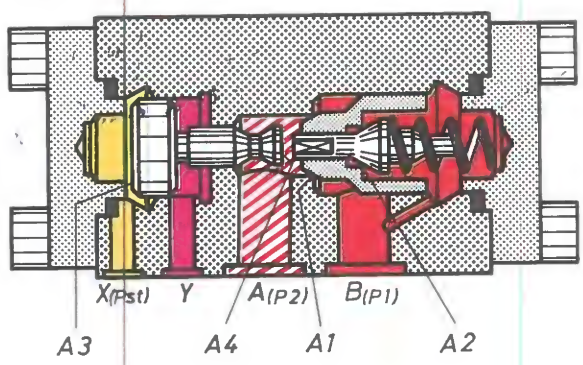
Fig. 5



It also shows that valve port A must be without pressure for pilot operation. Pressure at port A would work against control pressure at the control spool.

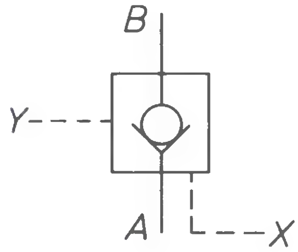
Model with drain port

Fig. 6



The sectional diagram shows valve type SL, with drain port and pilot opening poppet.

Symbol

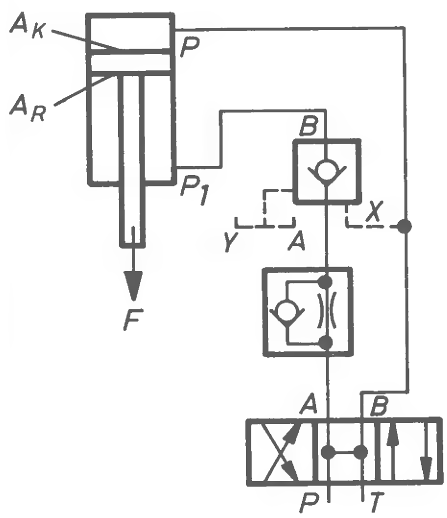


The difference between valve type SV is the additional drain port Y. The annulus area of the control spool is separated from port A. Pressure at port A affects only surface A_4 of the control spool (fig. 6).

Control pressure required at port X:

$$p_{St} = \frac{p_1 \cdot A_1 - p_2 \cdot (A_1 - A_4)}{A_3} + C$$

Fig. 7



The circuit shows that, with pilot operation, port A is pressurised by a pre-switched throttle valve (fig. 7)

In this case, a pilot operated check valve with external drain port is necessary.

Shut-Off Valves

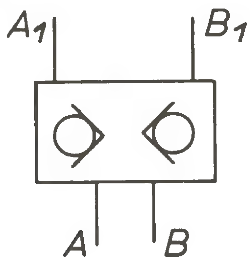
Important Technical Data:

size	10 to 150
flow	up to 6400 l/min
operating pressure	up to 315 bar

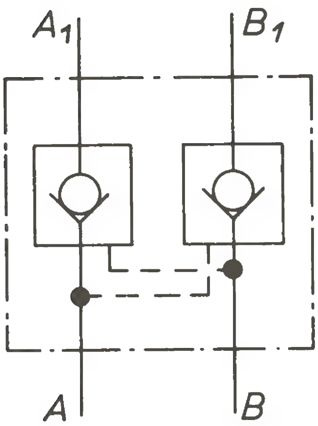
Double check valve
(double pilot operated check valve)

By fitting 2 pilot operated check valves 1 and 2 in one housing, a double check valve, type Z2S, is obtained (fig. 8).

Symbol simplified



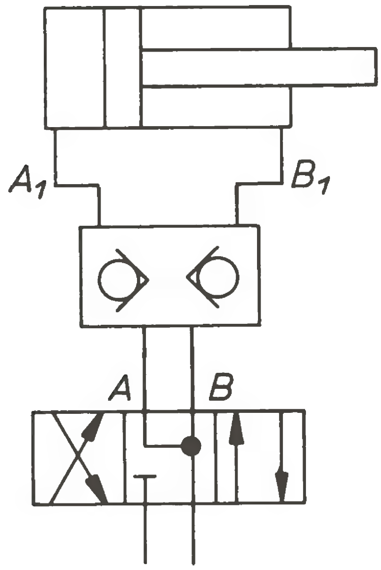
detailed



There is free flow in direction A to A_1, or B to B_1, while flow is blocked from A_1 to A or B_1 to B. If for example, there is flow from A to A_1, control spool 3 is pushed to the right and pushes the poppet of the check valve 2 from its seat.

The connection from B_1 to B is now open. The valve therefore operates at flow direction B to B_1.

The circuit example below shows the task of the double check valve:



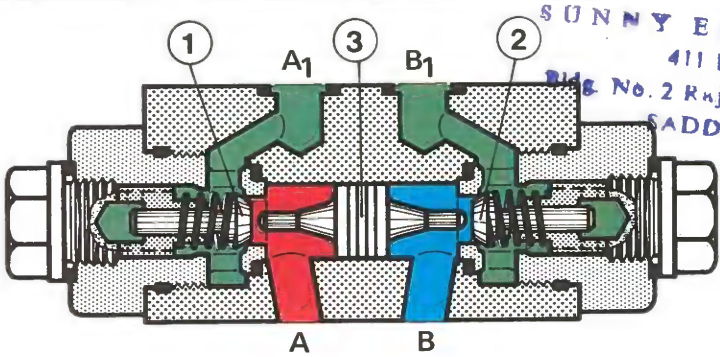
Both parts of the cylinder are closed leakfree. At any desired standstill position, the cylinder cannot be moved, even by an external force. This means, for example, that a cylinder under load will not begin to 'creep', even during a long standstill period.

In order to guarantee safe closing of both valve poppets, both user ports (A and B) must be unloaded at neutral position of the directional control valve by connection with the return line.

A double check valve is generally fitted as a sandwich plate between the directional control valve and the subplate.

Larger size valves are fitted with pilot opening poppets.

Fig. 8



Shut-Off Valves

Important technical data on the double check valve

Size:	6 – 25
Flow:	up to 300 l/min
Operating pressure:	up to 315 bar
Cracking pressure:	0.5 bar (sizes 6, 10) 1.0 bar (sizes 16, 25)

Prefill Valves

Hydraulically pilot operated prefill valves type SF, size 40 – 80

Prefill valves are large size hydraulic pilot operated check valves.

They are used mainly to prefill large cylinder volumes and to isolate the main working circuit under pressure, for example, in press construction.

In order to understand the fuction better, it can be explained using the sectional diagram in conjunction with the example of application shown (fig. 9 and 10)

Pilot poppet 1 and main poppet 2 are held on their seats by spring 3. Spring 4 pushes the control spool 5 into outlet position.

Port A is connected with a tank above the cylinder. Valve poppets 1 and 2 are affected by the oil above them. If the cylinder piston moves downwards (perhaps due to gravity when unloading the annulus area A_R or via fast forward cylinders), negative pressure occurs in the chamber above cylinder surface A_K. Related to the prefill valve, it also affects port B, i.e. on the rear of the closing poppet.

As it travels out, the cylinder sucks oil from the tank behind it, and thus opens the connection to the tank.

Fig. 9

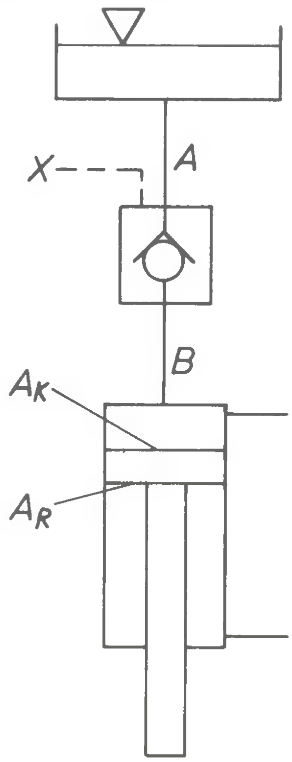
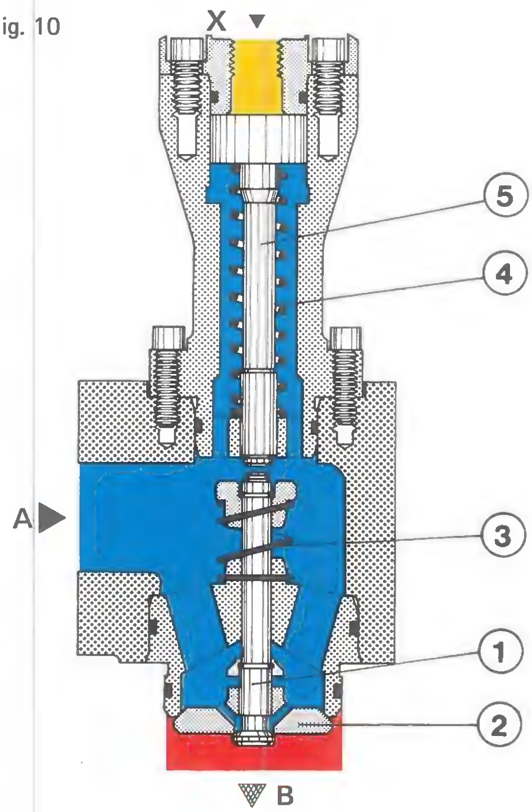


Fig. 10



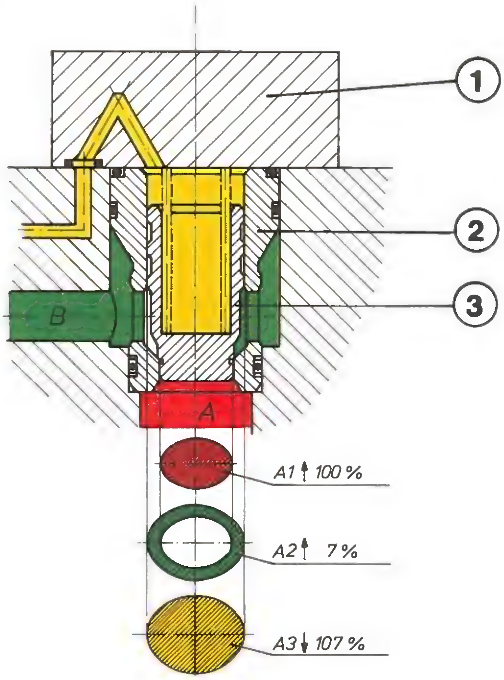
Shut-Off Valves

Shortly before the end of the zero stroke, the cylinder is braked to the desired speed. The pump now delivers oil into the cylinder chamber. The pressure now building up (working pressure) affects the rear of the valve poppet via prefill valve port B and thus isolates the working circuit from the tank.

After the working stroke, the cylinder should retract. By switching over the control elements, pressure affects, for example, the annulus area A_R (or the fast forward cylinder), and by means of prefill valve control port X, control spool 5. It pushes open the pilot opening poppet 1 and then the main poppet 2. The fluid above surface A_K can now be pushed back into the tank. The cylinder can travel inwards again.

Depending on application, the prefill valves can be fitted with or without pilot opening poppet.

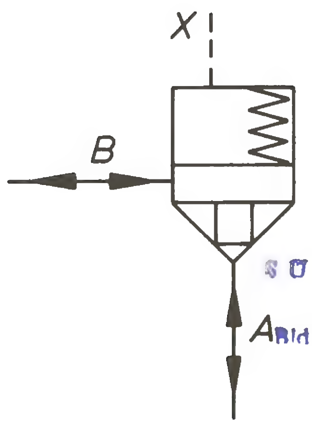
Fig. 11



2/2 Way Cartridge Valves (logic elements)

2/2 Way Cartridge Valves of different sizes

Symbol



2/2 way cartridge valves, often called "logic elements" comprise a cartridge assembly with a cover 1 and the control bores. The cartridge assembly is made up of a bush 2 with poppet surface and an opening or closing port 3 (called hereafter "valve poppet"), which is generally held on its seat by a spring (fig. 11).

Oil can flow through the valve from port A (on the bottom) to port B (on the side) or from B to A. According to the control, the cartridge valve is either open or blocked for the flow direction, i.e. the switching position is dependent only on the pressure conditions at this element.

The 2/2 way cartridge valve is therefore pressure dependent.

The valve poppet is graded, so that there are three important surfaces for the function.

The connections can be clearly explained using the sectional diagram.

Surface A_1 (at the seat) is regarded as 100%. The annulus area A_2, occurring due to the grading, is 7% or 50% of the surface A_1, depending on the model.

Shut-Off Valves

The area ratio $A_1 : A_2$ is therefore either 14.3 : 1 or 2 : 1. Surface A_3 is then $A_1 + A_2$ or can be 107% or 150% of surface A_1.

With reference to one size, the surface area A_3 is always the same. If the annulus area changes, then the surface area A_1, which is always regarded as 100%, also changes.

If surface A_3 is not affected by pressure (control port X without pressure), then the valve opens in both flow directions if the pressure force affecting the corresponding surfaces A_1 or A_2 is greater than the spring force.

If control pressure affects surface A_3, this pushes the valve poppet onto its seat, additionally to the spring force. The valve can thus open by unloading surface A_3 or by correspondingly high pressure at port A or B.

There are many possibilities for the control, i.e. influencing the force conditions at the valve poppet, due to the various models of cover and of the control drillings in the block.

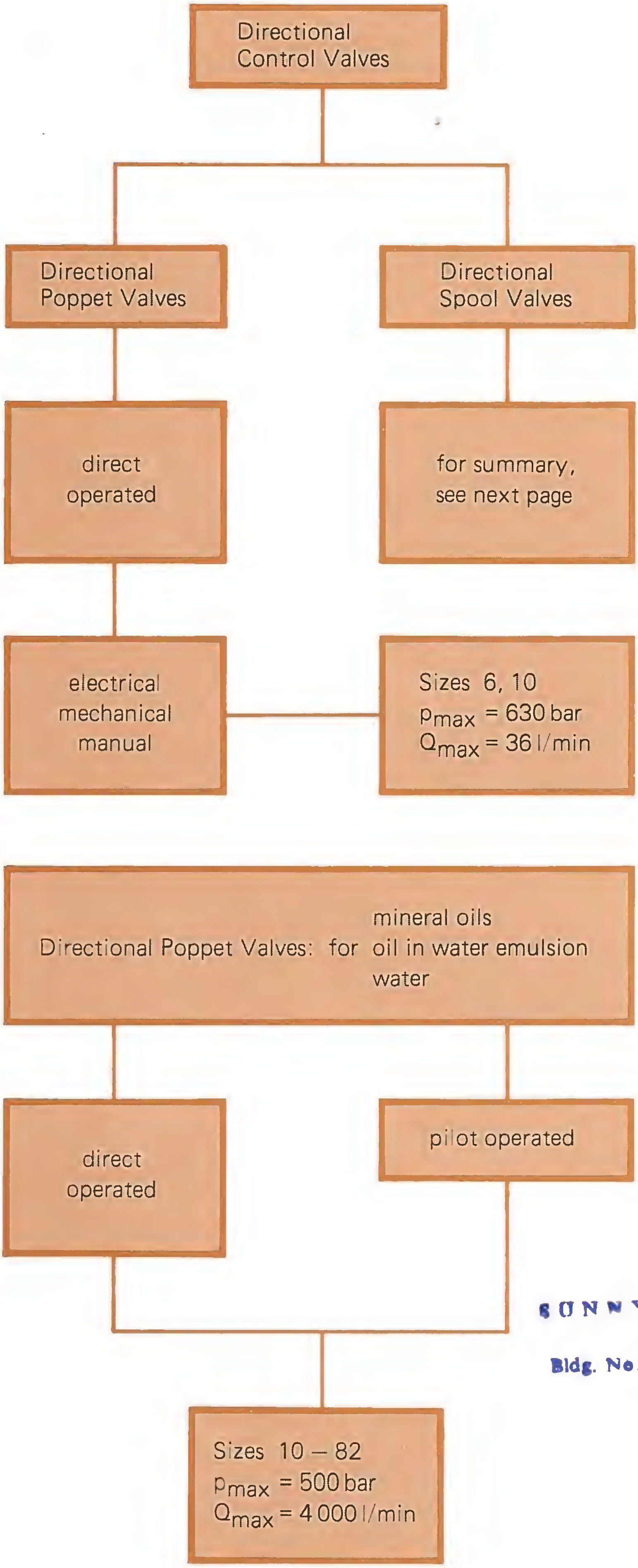
Notes on application

- The valve poppet is available with or without cushioning pin, whereby the model with cushioning pin is generally used together with the stroke limiter.
- If the 2/2 way cartridge valve is controlled from port B, it does not in general close so smoothly as when control is from port A. This refers in particular to the model with cushioning pin.
- The valve is leakfree from port B only with supply and drain of oil. If port A is used in the same way, leakage oil reaches B from A3 via the tolerance of the valve poppet guide.

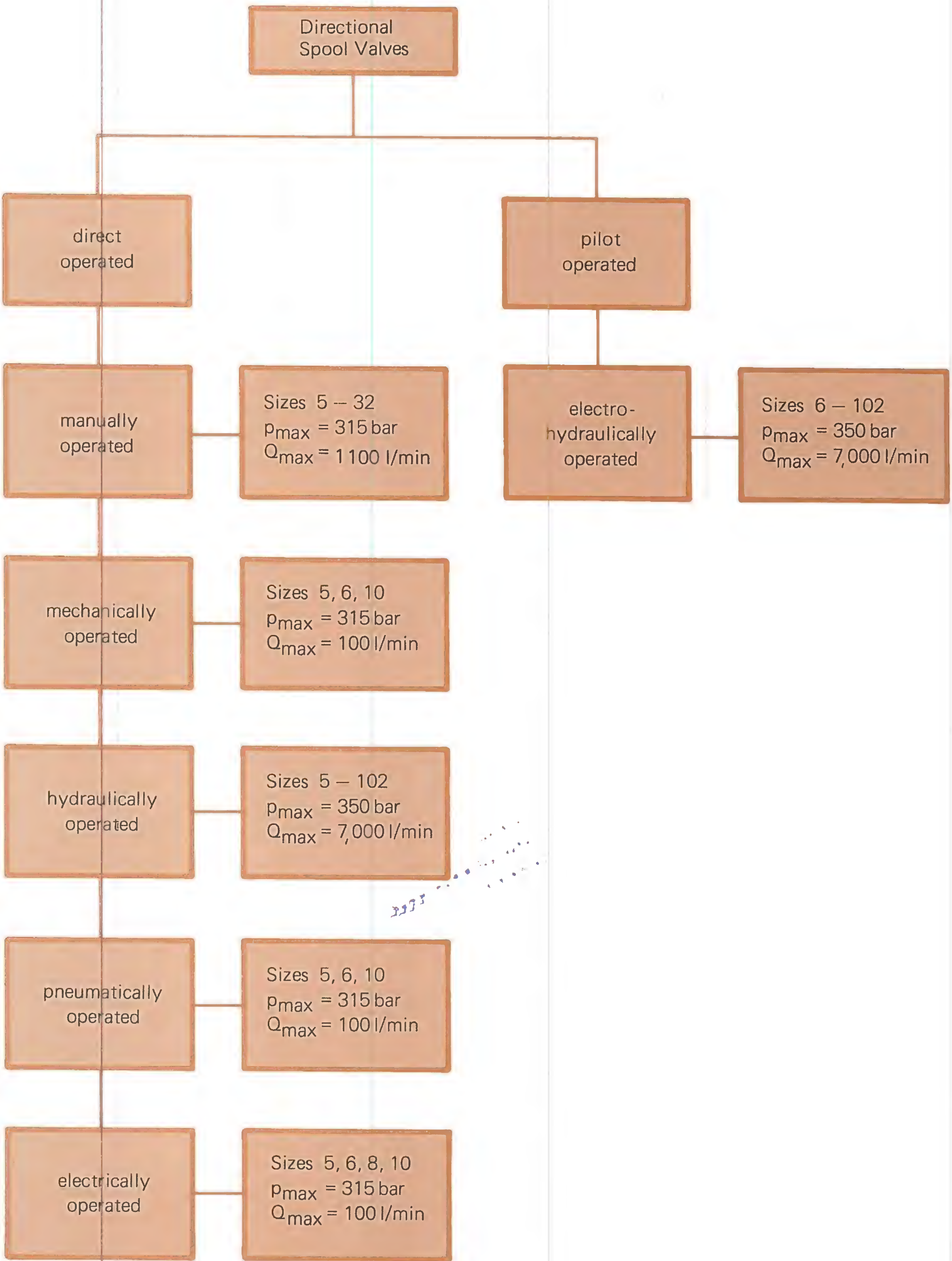
Where additional elements are used, other functions as, for example, pressure relief, throttling, or, by combination of several elements, multiple directional controls can be carried out.

Example of a control block with different types of cartridge valves

Programme Summary



Programme Summary



Directional Control Valves

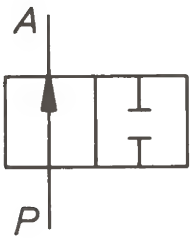
Purpose

Start, stop and direction of flow of a pressure fluid is controlled by means of a directional control valve, and thus the direction of movement or holding positions of a user (cylinder or hydraulic motor) is determined.

The designation of the directional valve refers to the number of service ports (control ports not included) and the number of switching positions.

A valve with 2 service ports and 2 switching positions is thus designated as a 2/2 way valve.

Symbol

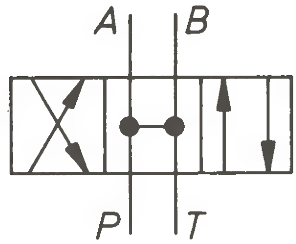


A valve with 4 service ports and 3 switching positions is then, in short, a 4/3 directional control valve.

P = pressure port (pump port)
T = tank
A, B = user ports

Directional Poppet Valves

Symbol



The ports are always designated at the outlet position.

These directional control valves can be sub-divided into 2 groups, according to their design.

1) directional poppet valves

2) directional spool valves

They can be either direct or indirect operated (pilot operated).
Whether a directional control valve is direct or indirect operated depends in the first place on the size of operating force required and thus on the constructional size (nominal size) of the valve.

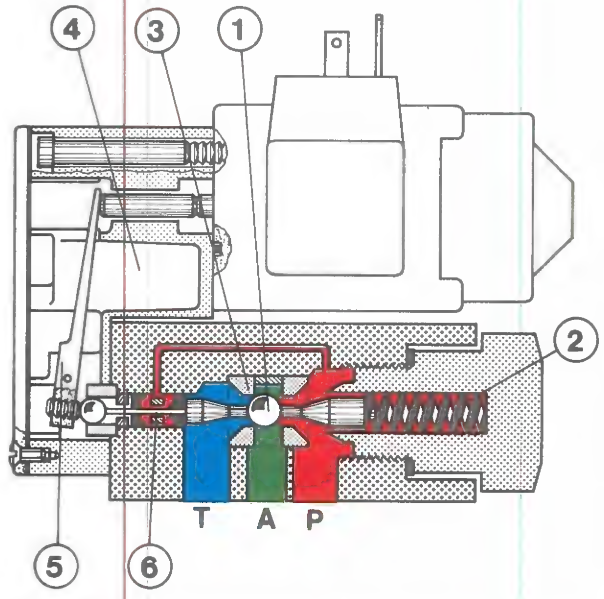
1) Directional Poppet Valves

Directional poppet valves differ mainly from directional spool valves by their leak-free closing, which cannot be achieved with spool valves, due to the tolerance required between spool and housing.

The photograph shows poppet valves with electrical and manual operation.

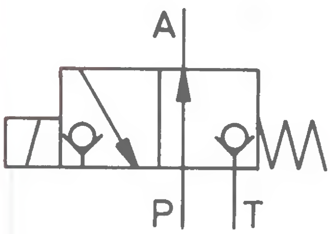
Directional Poppet Valves

Fig. 1



Direct electrically operated 3/2 way poppet valve, designed as one ball valve type SE ... U

Symbol



The seat element is a ball 1, which is pushed to the left onto the seat 3 (fig. 1) by a spring 2 in outlet position.

In outlet position, the connection from P to A is opened, port T is closed. The valve is switched by solenoid force or manual operation. The force affects the poppet element by means of a lever 5, with adjustment screw, ball and operating plunger, in the adapter 4. It is pushed to the right against spring 2 and pushed on its seat. Port P is now closed, and the connection A to T opened.

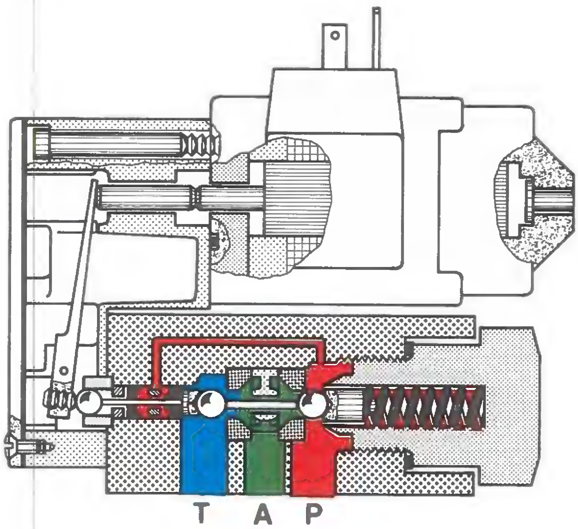
The operating plunger 6 is sealed in both directions with ring seals. The chamber between the two seals is connected to the P line. Pressure balance is thus achieved at the plunger, without having to overcome the pressure force affecting the poppet seat during switching. The valves can be used up to 630 bar.

During the switching process, the ports are connected to each other for a short period (see negative overlap).

The variety of flow configurations for spool valves is not possible for poppet valves. This is because of the special design of these valves.

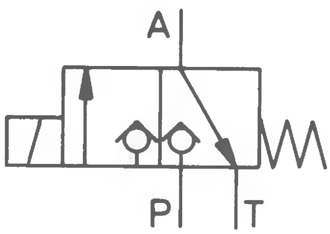
If you wish to exchange the two switching positions shown on the one ball valve, the 2 ball valve must be used (fig. 2).

Fig. 2



Direct electrically operated 2 ball valve type SE ... C

Symbol



On a 2 ball valve, the connection A to T is open and port B closed at outlet position. The spring and the pressure push the ball in the P line onto its seat. In switching position, the right ball is lifted from its seat, while the left ball is pushed on to its seat.

Valves with AC solenoids have a flexible lever and valves with DC solenoids a relatively rigid lever, corresponding to the different solenoid characteristics.

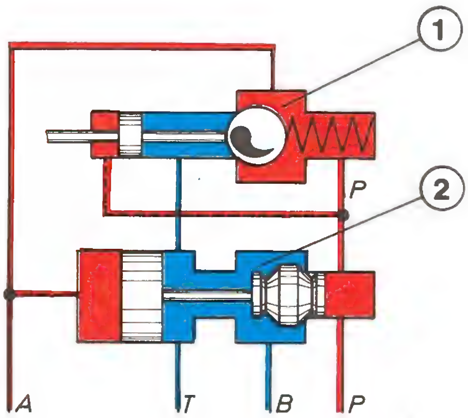
Notes

Directional Poppet Valves

Using a sandwich plate under a 3/2 way poppet valve, a 4/2 way valve can be obtained. The schematic diagram below shows the method of operation (figs. 3 and 4).

Outlet Position

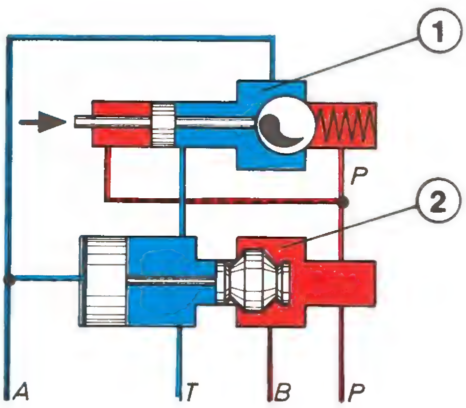
Fig. 3



The upper part represents the 3/2 way poppet valve 1, the lower part the sandwich plate 2.
In outlet position, the ball of 1 is raised off its seat. The connection from P to A is opened. A control line runs from A to the spool of valve 2. This surface is greater than that of the right seat element, which is therefore pushed to the right on to the seat. Port B in the sandwich plate is blocked from P and connected to T.

Switching position

Fig. 4



Directional Spool Valves

When operating a 3/2 way poppet valve, port P on top is closed. The connection from A to the tank is opened. At the same time, the large spool in the sandwich plate is unloaded.
Pressure at P pushes the spool with poppet element to the left and closes the connection from B to T. P is now connected with B and A with T.

Important technical data:

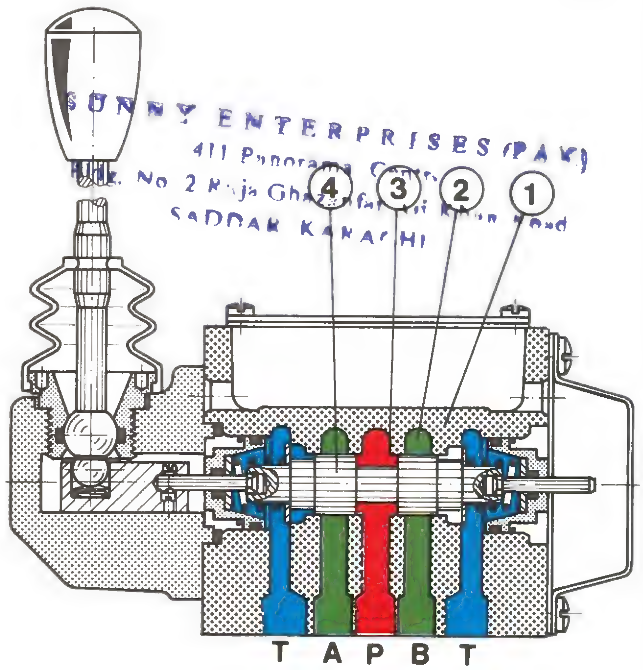
size	sizes 6 and 10
flow	up to 36 l/min
operating pressure	up to 630 bar

2) Directional Spool Valves

Directional spool valves can be sub-divided into linear spool and rotary spools.
The principle used most frequently is the linear spool valve, due to its many advantages:

Advantages:
- relatively simple design
- high switching power compared with rotary spool
- very good pressure balance and thus lower operating forces (see poppet valve)
- low losses
- many control functions.

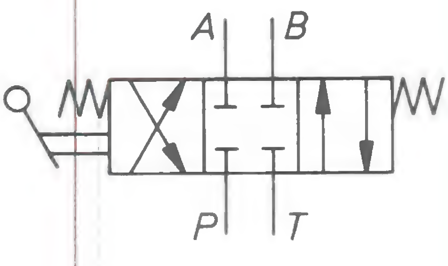
Fig. 5



4/3 way spool valve type WMM
Hand lever operated

Directional Spool Valves

Symbol



Construction (fig. 5)

Annular ports 2 are formed (generally cast) around a longitudinal bore in a housing 1. The longitudinal bore is interrupted by the annular ports. This results in control lands 3 in the housing. A moveable control spool 4 is in the longitudinal bore.

If the control spool is moved, it connects or divides the annular ports in the housing. Each annular port is connected to an outlet terminator in the housing. Separation and combination of the ports is synchronous. The operating sequence can be determined exactly.

The different control functions result relatively simply due to the spool shape. The housing does not generally change.

In our example, all ports P, T, A and B are separate at outlet position, i.e. without external operation. If the spool is now pushed to the right, for example, connections P to B and A to T occur.

Sealing for the individual annular ports to one another is achieved via the tolerance between the spool and the housing. Compared to poppet valves, hermetic sealing is not possible. The seal effect is therefore dependent on the dimensions of the sealing aperture and the fluid viscosity. This type of valve is unsuitable, for example, for water, but shows an adequate seal effect with oil.

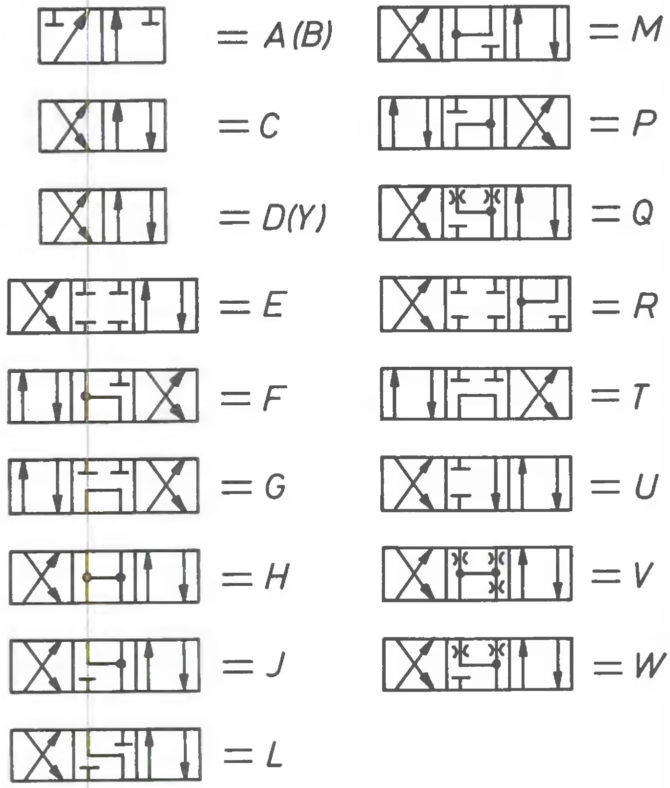
Switching Positions

The different switching functions are determined by the spool type and shown accordingly as a symbol. Each little box represents a switching position.

The following summary shows the most usual control functions, which are shown with letters.

The switching positions with the parallel and crossed arrows are the so-called "working positions".

Symbol



The position in the middle of 3 switching position valves is called the centre position.

For example, all ports can be closed (E), all ports can be connected (H), or else a combination is possible.

The centre position selected depends on the unit and the operation required.

Switch Overlap; Overlap Positions

When considering the switching function, the switching diagram which results when one switching position changes to the other, is also important.

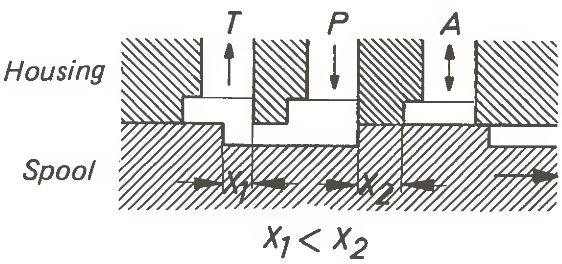
We differentiate between 3 switching overlaps:

1) Positive Overlap (fig. 6)

When the spool is switched to the right, the old connection P to T is blocked, before the new connection P to A is made. All ports are therefore separate from one another for a short time during the switching process.

Directional Spool Valves

Fig. 6

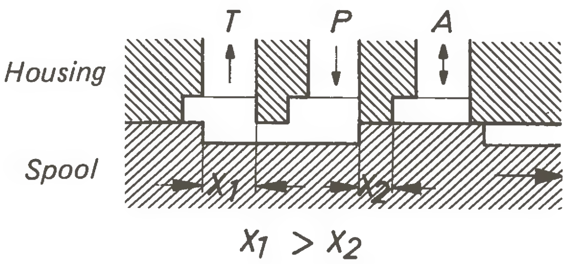


A pressure peak occurs, the degree of which depends on the switching time and the size of the oil flow.

A user under load cannot lower, or control pressure from the line in front of the valve is maintained.

2) Negative Overlap (fig. 7)

Fig. 7



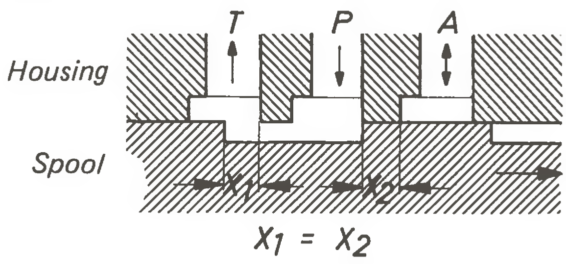
In this case, the connection to the user (port A) is opened during the switching process, before the connection to the tank is blocked. Thus all ports are connected for a short time during the switching process. This results in smooth switching. However, undesirable cylinder movements may occur at certain load conditions.

3) Zero Overlap (fig. 8)

The zero overlap lies between the overlap already described.

Direction $x_1 = x_2$. This means that the connection P to A is opened, as connection P to T is closed. This is generally used on servo valves, since a very small piston stroke should influence the oil flow.

Fig. 8

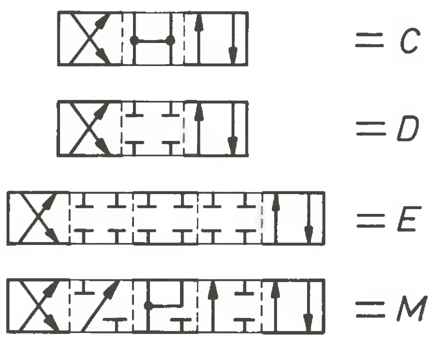


Representation of the Overlap Positions

As the overlap positions should also be known for the selection of the correct flow symbol, this is also shown on the detailed symbol representation.

Since this is not an actual switching position, the little box is shown by a dotted line.

Symbols



Directional Spool Valves

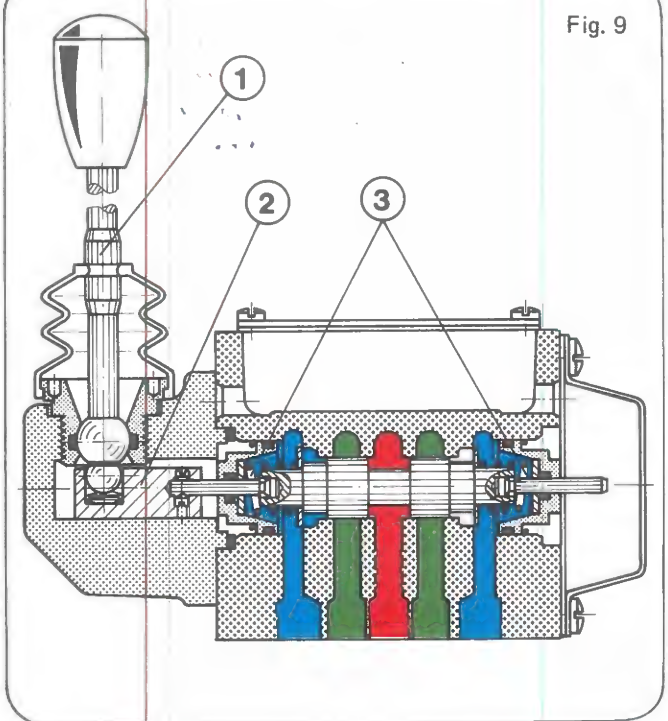
Direct operated directional spool valves

The control spool of a direct operated directional control valve is moved directly by a signal, without using an additional auxilliary force for switching. Direct operation can be carried out mechanically, hydraulically, pneumatically or electrically. The operating element is fitted to the end of the directional control valve housing.

Mechanical Operation

The photograph shows from left to right:
roller/lever operation type WMR
rotary knob operation type WMD
hand lever operation type WMM

Fig. 9



Hand lever operation type WMM

The sectional diagram (fig. 9) shows the operation of the control spool by means of hand lever 1.

The spool is fixed rigidly to the operating mechanism 2 and follows its movement.

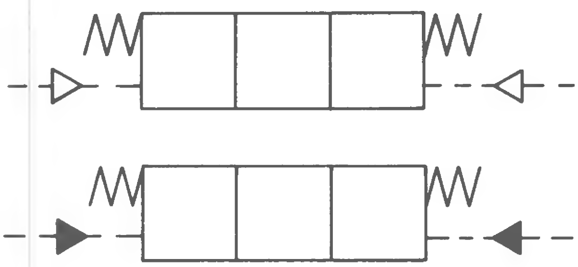
Return of the spool is by springs 3, which push the spool back into the off position, after the operating force stops (i.e. the hand lever is released). If a detent is fitted, the switching position is fixed with this and changed only by operating (not with roller/lever operation).

Hydraulic and Pneumatic Operation

The photograph shows hydraulic and pneumatically operated valves, sizes 5, 6 and 10 (from the right).

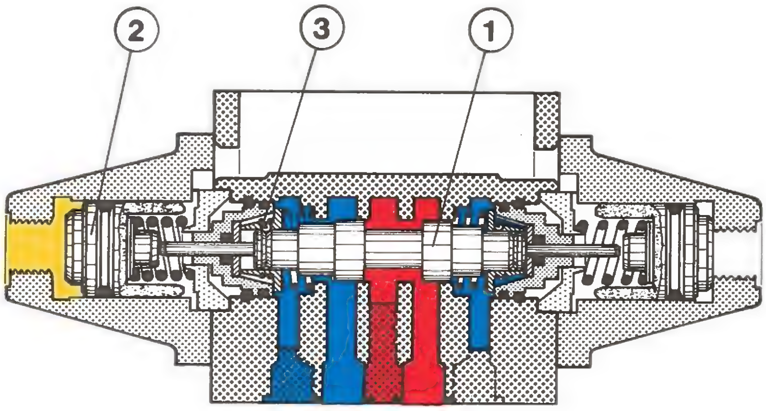
Symbol:

hydraulic and pneumatic operation with spring centering.



Directional Spool Valves

Fig. 10



The sectional diagram shows a valve with 2 switching positions (fig. 10).

Spool 1 is in the right switching position. This was achieved by pressurising the left operating cylinder. In this case, the switching position is fixed by means of a detent.

The control spool is not connected to the operating cylinder. Two operating cylinders are always necessary where there are two switching positions with detent or without spring return (impulse spool valve), also with 3 switching positions.

Where spool return with 2 switching positions is by means of springs, one operating cylinder is fitted.

Electrical Operation

Various models with solenoid operation

This type of operation is most frequently found, due to the use of large number of automatic processes. Four basic models of stroke solenoids are used:

- **Air gap DC solenoid**
  It is called a "dry" solenoid.
- **Oil immersed DC solenoid**
  It is known also a "wet" solenoid or "pressure-tight" solenoid.
  The solenoid armature runs in oil.
- **Air gap AC solenoid**
- **Oil immersed AC solenoid**

The **DC solenoid** has a high degree of reliability and gives a smooth switching process. It does not burn out, if it stops during a stroke, for instance, due to a sticking spool. It is suitable for high switching frequency

Directional Spool Valves

A characteristic of the **AC solenoid** is its short switching times. If the solenoid armature cannot switch through to its end position, the AC solenoid will burn out after a certain time (approx. 1 – 1 1/2 hours for oil immersed solenoids, approx. 10 – 15 minutes for air-gap solenoids).

Preference should be given to the **oil-immersed solenoid** for equipment in the open air or in a humid climate. Since the armature runs in oil, there is less wear, cushioned armature stroke and good heat transmission.

The **air gap solenoid** is the most simple design.

For the purpose of better comparsion, an air-gap AC solenoid 1 is shown left in fig. 11, and an air-gap DC solenoid 2 on the right.

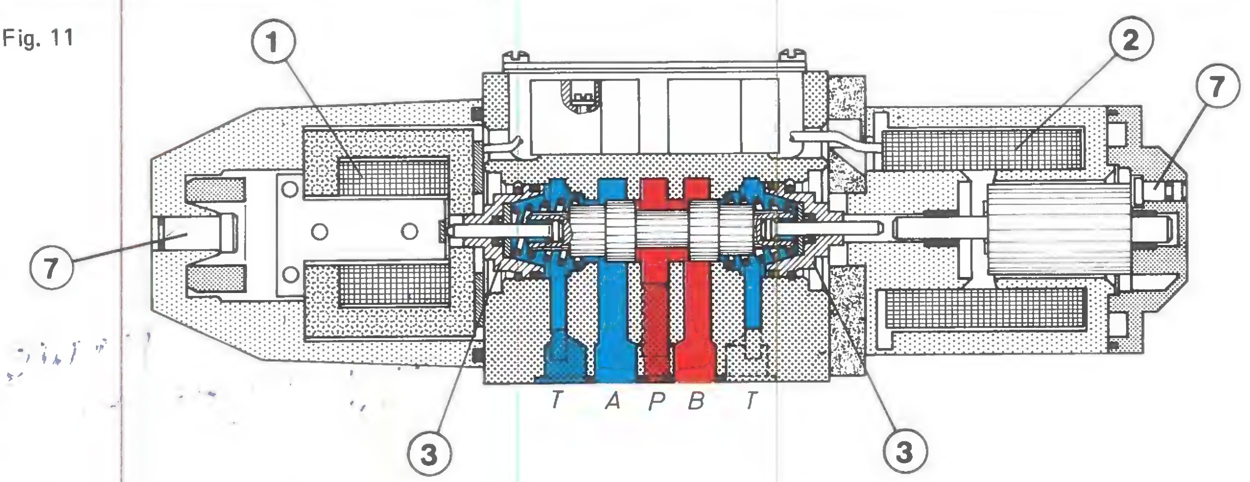
In this example, the valve has 2 switching positions, and the spool is not pushed into a certain position by springs.

When the solenoid coil is energised, the armature pushes the control spool by means of a plunger. In this case, AC solenoid 1 is operated and has pushed the spool to the right switching position.

On air-gap solenoids, the armature chamber is sealed to the tank line by seals in bushes 3.

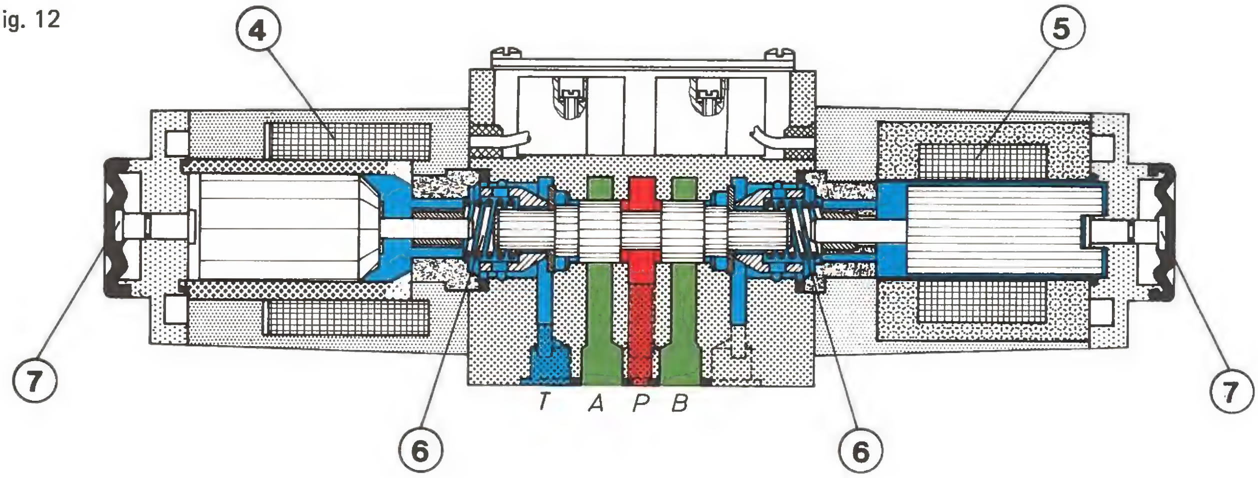
In this case, the springs serve only to fix the bush 3.

Fig. 11



Directional Spool Valves

Fig. 12



The sectional diagram (fig. 12) shows on the left hand side an oil immersed DC solenoid 4, and on the right hand side an oil immersed AC solenoid 5. The armature chamber is connected to the tank chamber. A valve with 3 switching positions is shown here.

The springs 6 are supported on the solenoid housings and centre the spool in neutral position by means of bush and disc.

Compared to the model with air-gap solenoids, the control spool is flat and is moved by means of the plunger at the solenoid armature.

The solenoids in both sectional diagrams are fitted with hand emergency 7. The control spool can thus be operated manually from outside. It is thus easy to check the switching function of a solenoid.

The directional spool valves shown up to now are designated as 3 chamber valves.

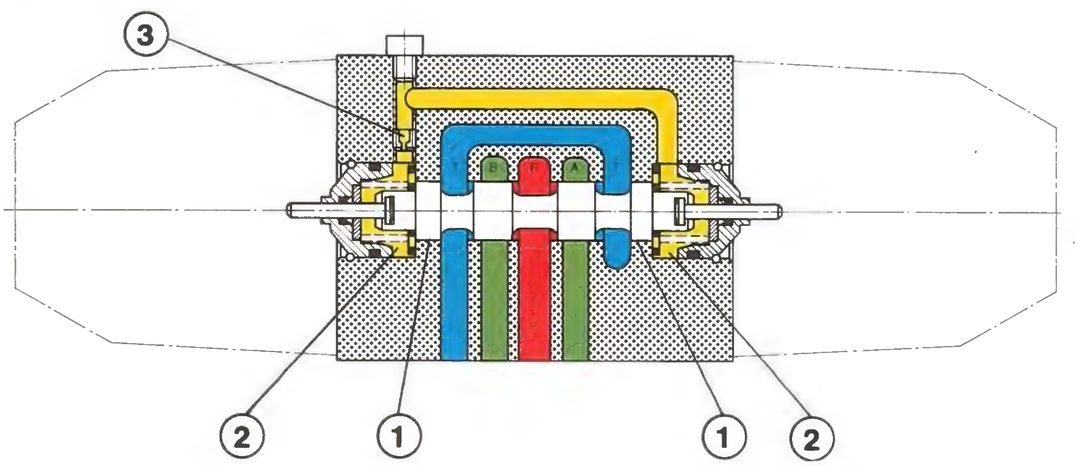
Lines P, A and B are separated by cross-pieces in the housing. The T line is not bocked, but has an external connection and is sealed by fitting the operating element or cover.

On a 5-chamber valve, the T line forms a chamber on each side by means of cross-piece 1 in the housing, as do P, A and B (fig. 13).

The two end chambers 2 are connected via a bore (yellow). If the control spool is moved, fluid is displaced from one end chamber into the other. A jet 3 or adjustable throttle can be fitted into this connection bore to influence the switching time corresponding to the diameter of the jet or the throttle position.

5 chamber valve

Fig. 13



Directional Spool Valves

Directional Spool Valves — important technical data

Operating Element	direct op.	pilot op.	Type	Q_{max} (l/min)	p_{max} (bar)	Size
Roller assembly	x		WMR	14	315	5
Roller assembly	x		WMR	30	315	6
Roller assembly	x		WMR	80	315	10
Hand lever	x		WMM	30	315	6
Hand lever	x		WMM	80	315	10
Hand lever	x		H - WMM	180	350	16
Hand lever	x		H - WMM	450	350	25
Hand lever	x		H - WMM	1100	350	32
Rotary knob	x		WMD	14	315	5
Rotary knob	x		WMD	30	315	6
Rotary knob	x		WMD	80	315	10
Pneumatic	x		WP	30	315	6
Pneumatic, p_{St} = 4.5 – 12 bar	x		WP	80	315	10
Pneumatic, p_{St} = 1.5 – 6 bar	x		WN	80	315	10
Hydraulic	x		WH	30	315	6
Hydraulic	x		WH	80	315	10
Hydraulic	x		H - WH	180	350	16
Hydraulic	x		H - WH	450	350	25
Hydraulic	x		H - WH	1100	350	32
DC and AC oil-immersed solenoids	x		WE	14	250	5
DC and AC oil-immersed solenoids	x		WE	60	315	6
DC oil-immersed solenoids	x		WE	100	210	10
AC oil-immersed solenoids	x		WE	100	210	10
DC air-gap solenoids	x		WE	100	315	10
AC air-gap solenoids	x		WE	100	315	10
DC oil-immersed solenoids, explosionproof	x		WE	30	60	6
DC oil-immersed solenoids, intrinsically safe		x	WEH	30	100	6
DC oil-immersed solenoids, explosionproof, explosionproof and sea-water resistant, explosionproof and weatherproof	x		WE	80	210	10
DC air-gap solenoids, with built-in limit switch	x		WE	80	315	10

Directional Spool Valves

Indirect operated (pilot operated) directional spool valves

Pilot operated, electro-hydraulic operated directional control valves for subplate mounting and flange connections.

Large size directional control valves, i.e. with high hydraulic power (= p x Q) are pilot operated.

The reason for this is the operating force necessary to push the control spool and the size of solenoids required for this.

For this reason, directional control valves up to size 10 are direct operated, over size 10 pilot operated. Exceptions to this are directional control valves with hand lever up to size 32, where the levers have corresponding dimensions.

A pilot operated directional control valve comprises the main valve 1 and the pilot valve 2 (fig. 14).

The pilot valve is generally direct operated electrically (solenoids). When the pilot valve is switched, the control signal is amplified hydraulically and the main control spool moved.

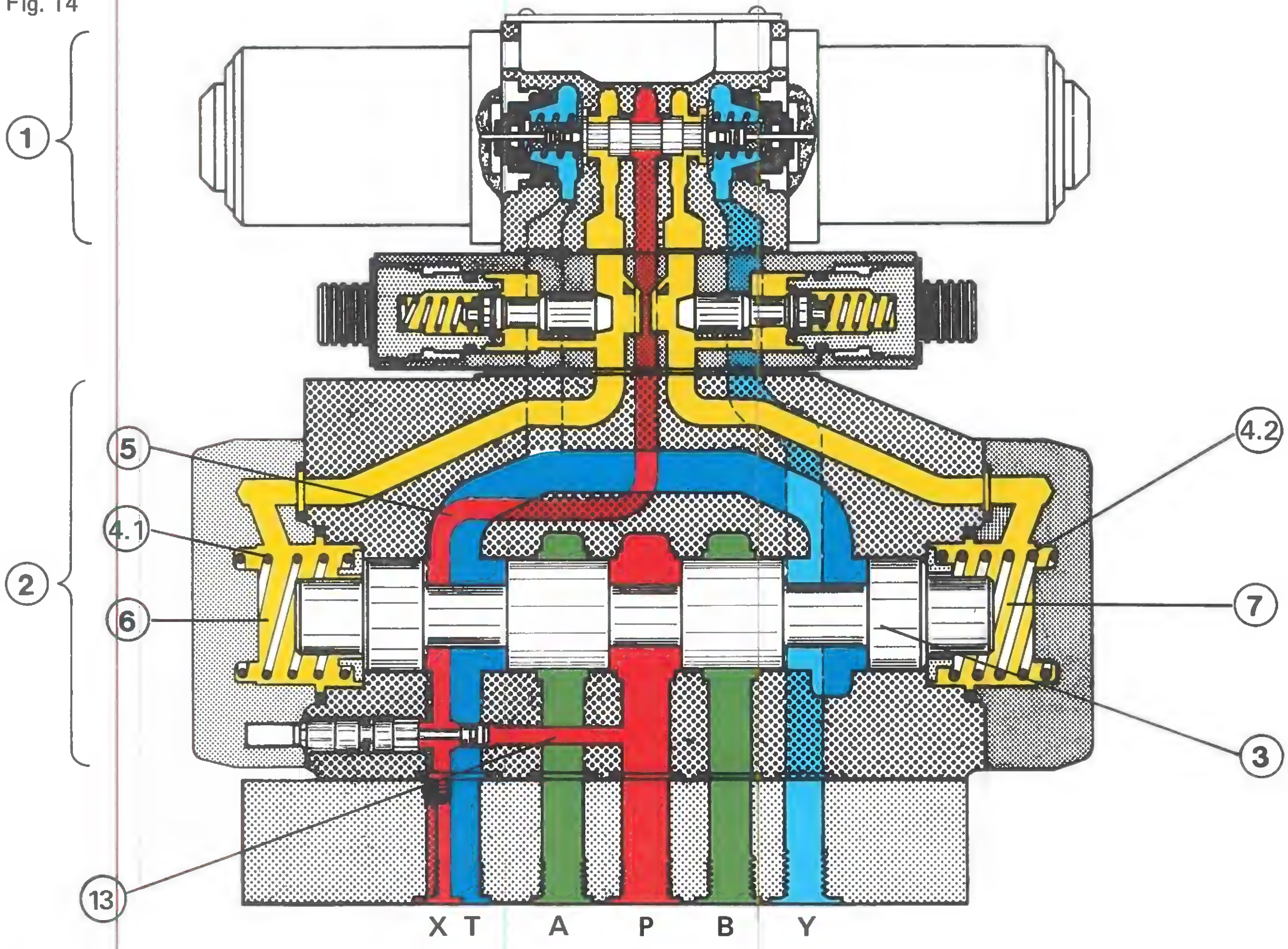
On size 102 (up to 7000 l/min) the pilot valve is itself a pilot operated valve.

This is not due to the operating forces, but to the amount of pilot oil required.

Directional Spool Valves

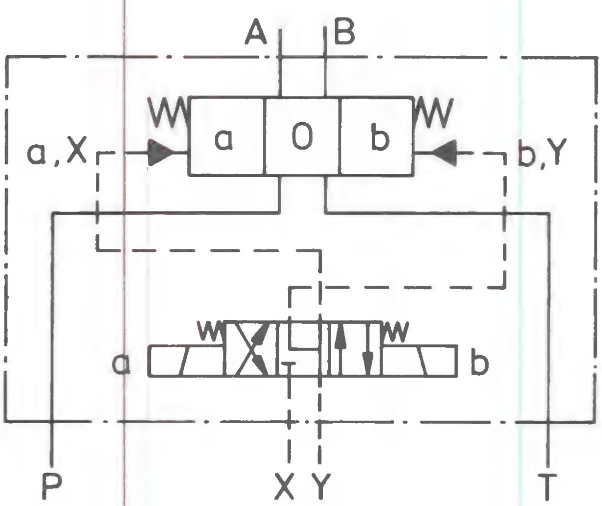
Electro-hydraulic operated directional control valve spring centered

Fig. 14

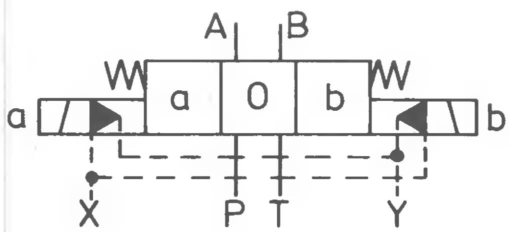


Symbol, detailed

x = external y = external (for example)



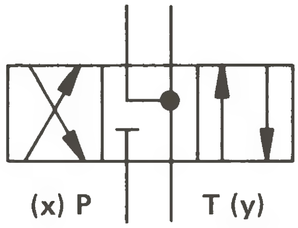
simplified



The pilot valve is an electrically direct operated 4/3 directional control valve (fig. 14).

On the spring centered model, the main control spool 3 is held in centre position by the springs 4. Both spring chambers (yellow) are thus connected in neutral position via the pilot valve with the tank (light blue) without pressure. Thus the centre position for the pilot valve is fixed.

Directional Spool Valves



Oil is supplied to the pilot valve via control line 5. Supply is either **internal** or **external** (for exact details see page 107).

If, for instance, the right-hand solenoid at the pilot valve is operated, this moves the pilot valve spool to the left.

The left-hand spring chamber 6 is thus subjected to pilot pressure, the right-hand spring chamber 7 remains unloaded to tank.

Pilot pressure acts on the left surface of the main spool and pushes it to the right against the spring 4.2, until it reaches the cover. Ports P and B and A and T in the main valve are thus connected. When the solenoid is shut off, the pilot valve returns to centre position and the spring chamber 6 is unloaded to tank again. The spring 4.2 can now push the main spool to the left, until it touches the retainer of spring 4.1. The spool is in centre position (neutral position).

The control oil from spring chamber 6 is pushed into the y line via the pilot valve. Pilot drain can also be external or internal, as can the supply.

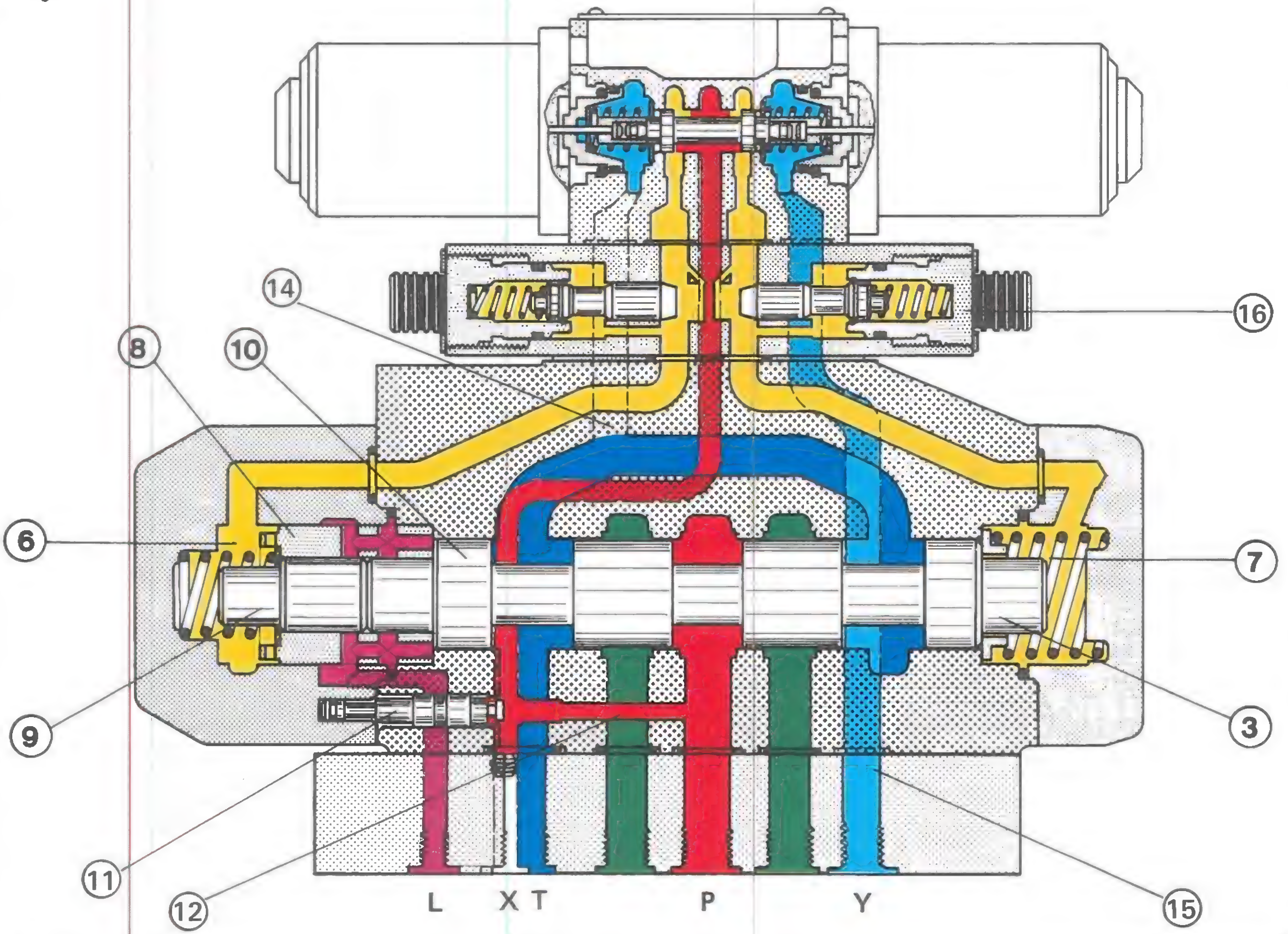
The switching prcedure for the left switching position is similar.

A certain minimum pilot pressure is necessary to operate the main control spool, according to symbol and valve model.

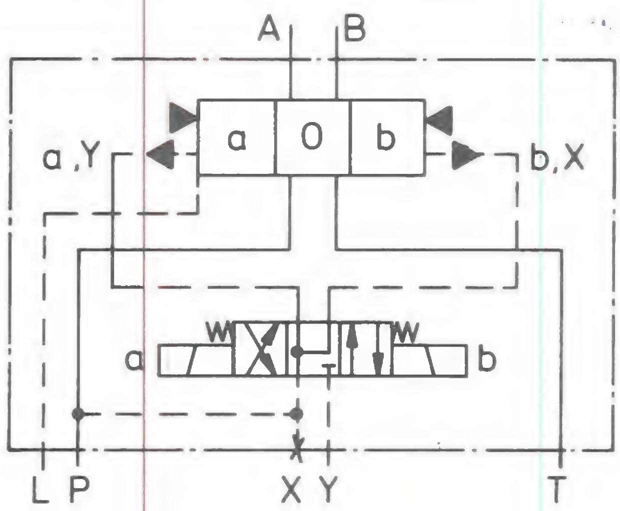
Directional Spool Valves

Electro-hydraulic operated directional control valve pressure centered

Fig. 15



Symbol, detailed
x = internal, y = external (for example)



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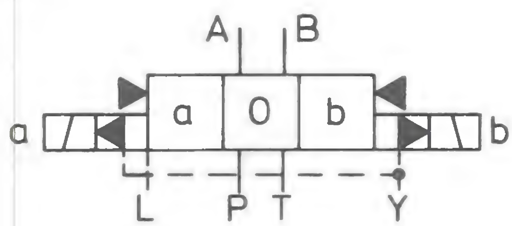
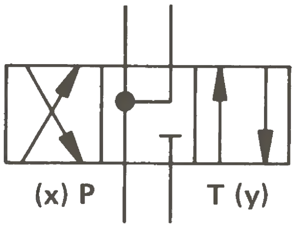


In centre position on pressure centered valves, both control chambers 6 and 8 are connected with control pressure. The main control spool is held in centre position by the effect of the pressurised surfaces of spool 3, centering bush 8 and centering bolt 9.

Directional Spool Valves

Centre position of pilot valves



If the right-hand solenoid at the pilot valve is operated, this moves the pilot spool to the left. The control chamber 6 therefore remains connected with the control pressure, while control chamber 7 is unloaded to tank. The centering bush 8 touches the housing. The centering bolt 9 pushes the main spool control spool to the right until it reaches the stop.

The springs in chambers 6 and 7 serve, for instance, to hold the spool in centre position, even with horizontal valve arrangement and without pilot pressure.

When the right-hand solenoid is de-energised, the pilot spool returns to centre position and control chamber 7 is connected with control pressure again.

The spool surface 3 is larger than the surface of the centering pin 9. The main spool moves to the left until the full diameter land touches the centering bush. The surfaces of the centering bush and pin are larger than the spool area 3; the spool remains in centre position.

If the left-hand solenoid is operated, then the pilot spool moves to the right. Control chamber 7 remains connected with control pressure, while control chamber 6 is unloaded to tank. Surface 3 is under pressure, causing the main control spool to move to the left, until it touches the centering pin 9 at the cover. The centering bush 8 is also moved.

The switching position in the main valve is reached when the left-hand solenoid is de-energised, the pilot spool returns to centre position and control chamber 6 is connected with the control pressure again.

The surfaces of centering bush 8 and pin 9 under pressure are larger than the spool surface 3.

The main spool moves to the right until it touches the centering bush at the housing. The spool surface 3 on the right side is now greater than the surface of centering pin 9 acting on the left side, and the spool remains in centre position.

A drain port (violet) is necessary to unload pressure in the chamber between the main spool and the centering bush.

Internal Pilot Supply (fig. 15, item 12)

The control fluid in the main valve is taken from the P line and fed to the pilot valve via the control line (red). This is represented as an example in the sectional diagram "pressure centered directional control valve".

Control port x must be closed and the sealing pin 11 mounted as shown.

No separate pilot circuit is required for internal pilot drain. However, one or two points must be taken into consideration for practical application:

- If the main control spool has negative overlap or bypass flow in centre position, the required pilot pressure does not build up or else breaks down during the switching process. A pressurising valve, for instance, would have to be fitted in the P line. The valve does not open until the minimum control pressure has built up. The connecting bore to the control line branches off in front of the pressurising valve. However, with this solution, there is continuous loss of pressure (for pressurising valve, see accessories page 108).
- An excessive quantity of oil can flow to the pilot valve, for instance, through an accumulator, and this can cause damage.
- Care must also be taken that the operating pressure does not exceed the maximum control pressure, otherwise a pressure reducing valve must be fitted. It brings about a reduction of the pilot pressure; for the valves described this is in the ratio 1 : 0.66.
  At the same time, care must also be taken that pressure does not fall below the required minimum.

External Pilot Supply (fig. 14, item 13)

The pilot oil is taken from a separate control circuit, which in any case can be better adapted to the requirements of pressure and flow, than with internal supply.

On the valves shown, it is easy to change "internal to external" or vice versa, by changing the mounting position of the sealing pin 11. For modification purposes, it is necessary only to dismantle the cover and move the sealing pin 11.

The correct mounting position for external pilot supply is shown in the sectional diagram "spring centered directional control valve". The sealing pin separates the connection of the control line from the P line.

Directional Spool Valves

Internal Pilot Drain (fig. 15, item 14)

Oil flowing back from the pilot valve is fed direct into the T line of the main valve. It must also be borne in mind that pressure surges occuring in the T line when switching the main control spool affect the unloaded control chamber via the pilot valve.

External Pilot Drain (fig. 15, item 15)

Oil flowing back from the pilot valve is not fed into the T line of the main valve, but fed separately back to tank via port y.

The sectional diagram (fig. 15) shows internal pilot drain and external pilot drain at the same time, for comparsion purposes.

Pilot Choke Adjustment (fig. 15, item 16)

The pilot choke adjustment is also shown in the sectional diagram. It is designed as a sandwich plate and can be fitted between the pilot valve and the main valve. This is a double throttle/check valve 16.

According to mounting position, the oil flowing to or from the control chambers is throttled and thus the switching time of the main spool influenced.

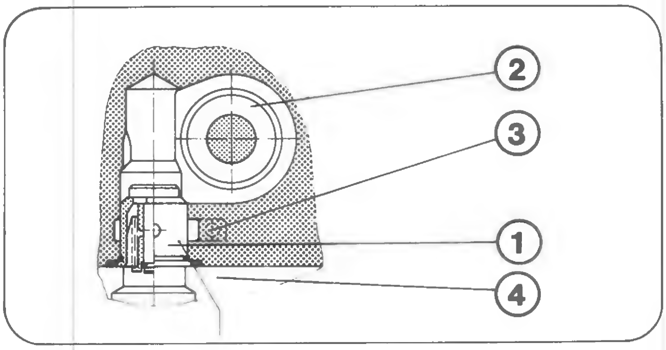
With the mounting position shown, the oil drain is throttled. In the supply line, the check valve is open.

Accessories

A stroke adjustment can be fitted. This means that the main spool can carry out only a limited stroke. An approximate throttling of the main flow can thus be achieved for the flow direction. It is possible to observe the position of the main spool by fitting an end position indicator.

Pressurising Valve

It is necessary to fit a pressurising valve (check valve) in the P line of the main valve on valves with bypass flow and internal pilot supply, in order to build up the minimum control pressure.



1 Pressurising valve
2 P line in valve housing
3 X line
4 Subplate

Directional Spool Valves — important technical data

Operating Element	direct op.	pilot op.	Type	Q_{max} (l/min)	p_{max} (bar)	Size
Electro-hydraulic		x	WEH	160	315	10
Electro-hydraulic		x	WEH	300	350	16
Electro-hydraulic		x	WEH	650	350	25
Electro-hydraulic		x	WEH	1100	350	32
Hydraulic, electro-hydraulic	x	x	WH/WEH	2000	350	52
Hydraulic, electro-hydraulic	x	x	WH/WEH	3000	350	62
Hydraulic, electro-hydraulic	x	x	WH/WEH	4500	350	82
Hydraulic, electro-hydraulic	x	x	WH/WEH	7000	350	102

Notes

Programme Summary

Pressure Control Valves

Pressure Control Valves	Operation	Valve type	Sizes	p_{max}	Q_{max}
Relief	direct operated	Poppet Valve	6 to 30	630 bar	330 l/min
Relief	pilot operated	Poppet Valve, Spool Valve	8 to 82	315 bar	3500 l/min
Sequencing	direct operated	Spool Valve	5, 6, 10	210 bar	45 l/min
Sequencing; Accumulator charging	pilot operated	Spool Valve	10, 25, 32	210 bar	450 l/min
Reducing	direct operated	Spool Valve	5, 6, 10	210 bar	45 l/min
Reducing	pilot operated	Poppet Valve, Spool Valve	8 to 32	315 bar	300 l/min

Pressure Control Valves

Pressure control valves serve to influence pressure in a unit or in part of a unit.

The valves can be sub-divided into 3 groups, according to function:

1. pressure relief valves
2. pressure sequence valves (accumulator charging valves)
3. pressure reducing valves

The valves can be direct or pilot operated.

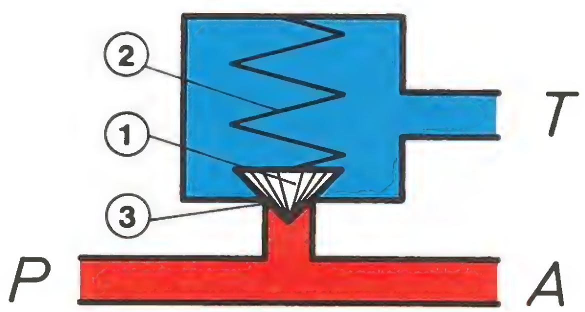
The photograph shows pressure relief valves, direct and pilot operated

1. Pressure Relief Valves

Direct Operated Pressure Relief Valves

Firstly, we shall look in general at the basic principle of the direct operated pressure control valve.

Fig. 1

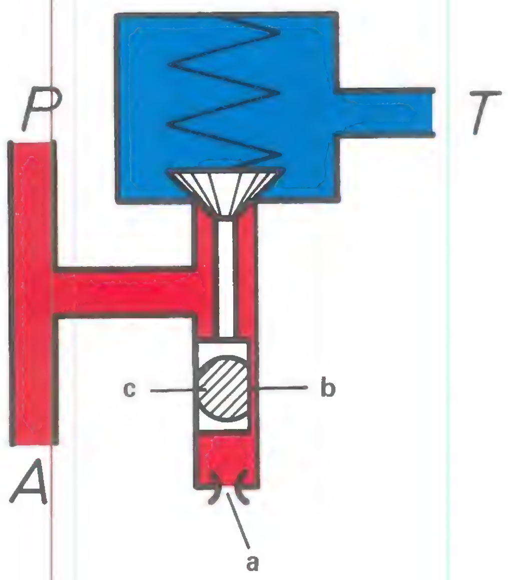


A closing element 1 is pushed on its seat 3 with a certain force by spring 2, according to its size and pre-tension. The spring chamber is unloaded to tank. Pressure in the system acts on the surface of the closing element. Pressure multiplied by surface area results in a force, which works against the spring force. This force also increases as pressure increases. As long as the spring force is greater than the pressure force, the closing element remains on its seat. If the pressure force exceeds the spring force, the element is pushed against the spring and opens the connection. The excess fluid flows back to the tank. When the fluid drains via the pressure control valve, the hydraulic energy is transformed into heat. The heat quantity $W = \Delta p \cdot Q \cdot \Delta t$.

If, for example, no fluid is taken by the user, the total quantity should drain via the valve. The valve opens, until there is balance between the pressure and spring force at the closing element. The opening stroke changes all the time with the flow supplied. The pressure value set according to the spring force is not exceeded. The valve is also described as a safety valve.

Pressure Control Valves

Fig. 2



On the previous drawing, only the forces, i.e. the static side of the valve was considered. Seen from the dynamic side, we have a spring -weight-system, which causes oscillations when it moves. These vibrations affect the pressure, and must be eliminated by cushioning.

Possibilities for cushioning are, for example: (fig. 2)

a) cushioning spool and jet to the spool chamber

b) cushioning spool with one surface

c) cushioning spool with correspondingly large tolerance play (cushioning groove).

The spool is fixed rigidly to the closing element. At spool stroke movement, the fluid must be fed via the jet or the cushioning groove. A cushioning force occurs opposite to the direction of movement.

Symbol
direct operated pressure relief valve

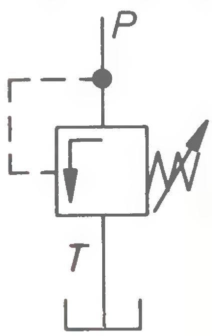
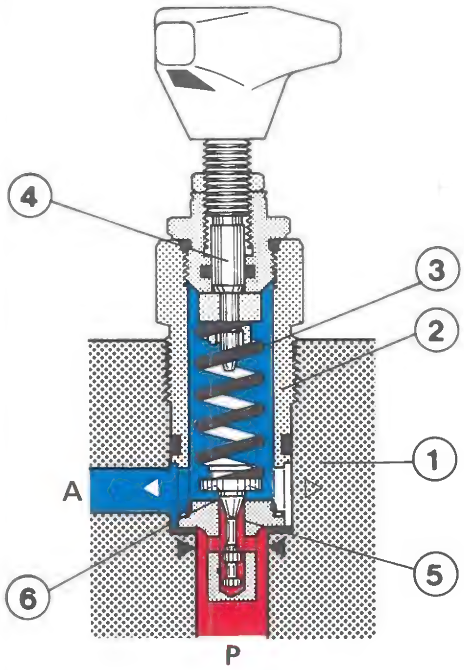


Fig. 3



Direct operated pressure relief valve type DBD, as cartridge valve

The housing or control block 1 contains the sleeve 2, spring 3, advance mechanism 4, poppet with cushioning spool 5 and, as a separate part, the hardened seat 6.

The spring pushes the poppet in its seat. The spring force can be steplessly adjusted by means of the rotary knob. The pressure is thus also set accordingly. Port P (red) is connected with the system. Pressure in the system acts on the poppet surface. If pressure lifts the poppet from its seat, the connection to the T port (blue) is opened. The poppet stroke is limited by a pin.

Since the spring force also increases according to the spring constant as the stroke increases, the underside of the spring retainer is a special shape. The impulse forces of the oil flow are used in such a way that the increase in spring force is almost balanced out.

In order to maintain good pressure setting and a flat p-Q curve over the complete pressure range (if possible, low pressure increase with increasing flow), the total pressure range was sub-divided into stages. One pressure stage corresponds to a certain spring for a maximum set operating pressure.

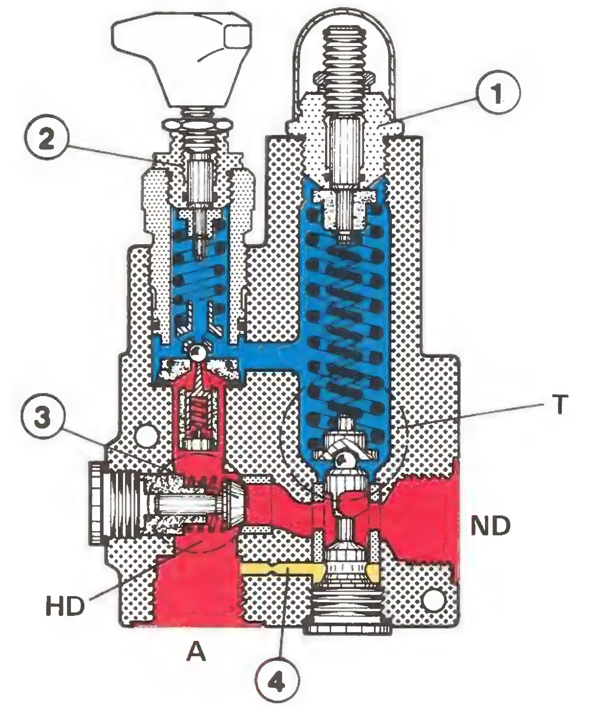
Pressure Control Valves

Important Technical Data:

size	sizes 6 – 30
flow	up to 330 l/min
operating pressure	up to 630 bar

Two Stage Valve Type DU

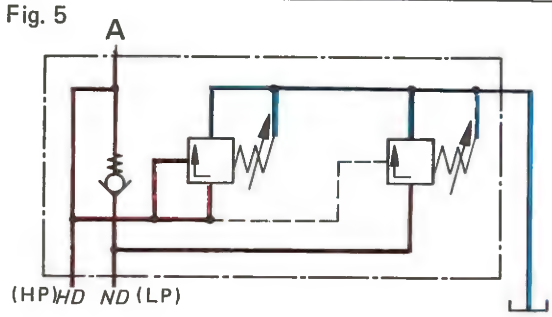
Fig. 4



The two-stage valve represents a combination of two direct operated pressure relief valves. It comprises pressure control valve 1 for the low pressure side, pressure control valve 2 for the high pressure side and a check valve 3. The valve is used to control a pump combination for high pressure (HP) and low pressure (LP). Where there is low working resistance, for instance, on a cylinder, both pumps, HP and LP, deliver oil into the system (port A). The HP pump has a low delivery compared to the LP pump. Supply of oil from the low pressure pump is via the check valve 3.

The user travels out at high speed. If pressure in the system increases to the value set at valve 1, it is piloted open from the HP side via control line 4. The low pressure pump now delivers to the tank almost without pressure. The check valve 3 blocks the HP circuit from the LP circuit. There is also a leak-free block between the control line 4 (yellow) and the LP side.

Fig. 5



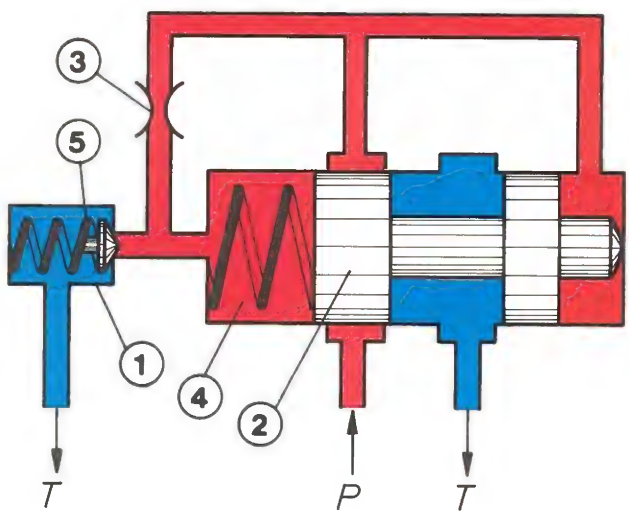
The HP pump now delivers oil into the system. The user moves at low speed. System pressure can rise to operating pressure value. Maximum operating pressure and thus the fuse protection of the high pressure circuit, is set at valve 2. When maximum pressure is reached, valve 2 opens the connection to the tank port.

Pilot operated Pressure Relief Valve

Pilot operated pressure relief valves are used for larger flows.

Here again we shall look firstly at the basic principle:

Fig. 6



The pilot operated pressure control valve comprises a pilot valve 1 and the main valve 2. The pilot valve is a direct operated pressure control valve.

The pressure in the system (red) affects the right-hand side via port P, the left-hand side of the main spool via a jet 3, and also the poppet of the pilot

Pressure Control Valves

valve. At standstill, the pressure is equal on both sides of the spool. As the surfaces are of equal size, the spool is pressure-balanced. The spring 4 holds the main spool in the starting position shown. P and T are separated from each other. The response pressure of the pilot valve is set at spring 5 of the pilot valves.

If system pressure reaches the value set at the pilot valve, the oil flows to tank via the jet and the pilot poppet. A pressure drop occurs at the jet, which is also effective between the two main spool surfaces. If the force of pressure drop x spool surface exceeds the spring force 4, the main spool is pushed to the left, allowing the excess oil to drain out of the system to the tank.

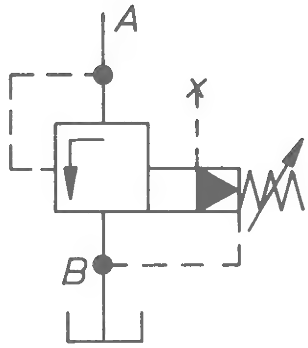
Fig. 7 shows a practical model of the pilot operated pressure relief valve. Input pressure affects the pilot valve 1 by means of a jet combination (3.1 and 3.2) in the control line, and also the spring loaded side of the main spool by means of a further jet (3.3).

During the response time of the pressure control valve, the control oil flows continuously by means of the spring chamber back into the tank. A filter mesh 6 is provided to protect the jet (3.2) from particles of dirt.

The jet (3.3) serves to cushion the main spool.

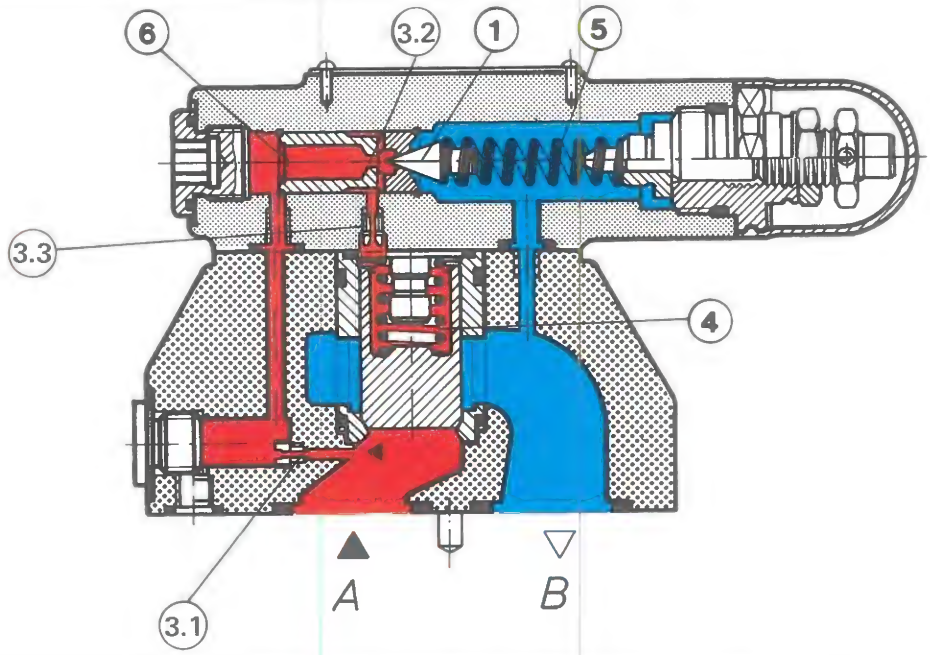
The spring 4 is relatively weak, so that the increase of spring force is of no importance with a greater opening stroke.

Symbol



The control fluid can drain either internally (as shown) or externally. It must be taken into account that the peak pressure in the drain line with internal pilot drain affects the spring side of the pilot valve. The response pressure increases by the value of the pressure peak.

Fig. 7



Pilot operated pressure relief valve, type DB

Pressure Control Valves

Pilot Operated Relief Valve with directional valve unloading

Fig. 8

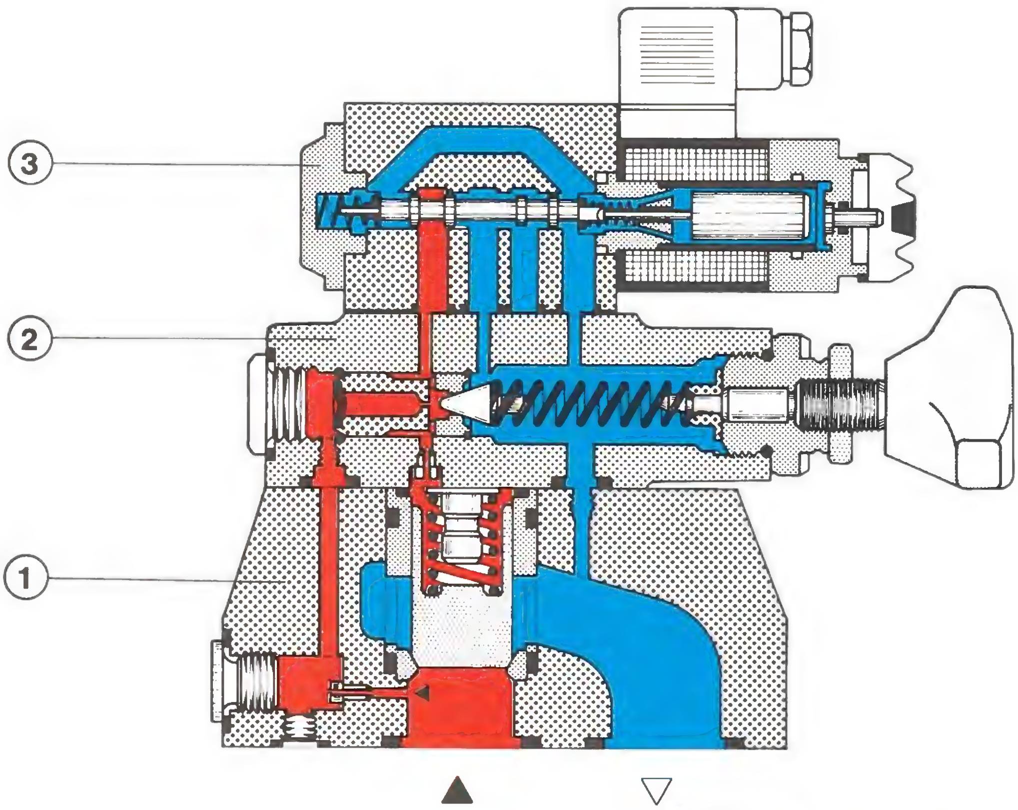
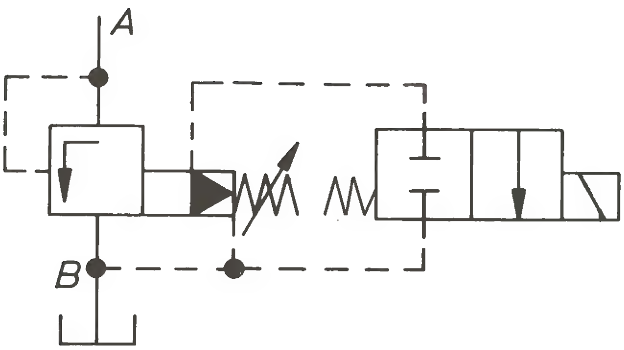


Fig. 8
1. main valve
2. pilot valve
3. directional valve

Symbol



The valve shown previously has now been enlarged by a direct mounted (2/2) directional control valve.

In the position shown, the directional valve blocks the line (red) which is connected in front of the pilot poppet. The pressure control valve is as already described.

In switched position, the directional valve connects the spring side of the main spool with the tank. This side of the main spool is thus unloaded and the fluid can drain to the tank almost without pressure against the spring force (approx. 3 bar).

The combination with the directional valve makes it relatively easy to achieve by means of a control signal bypass flow, almost without pressure, via the pressure control valve.

Applications would be, for instance, starting a pump without pressure, or flow circulation without pressure during unit standstill periods and thus low power loss.

Important Technical Data of Pilot Operated Relief Valves

Sizes	sizes 8 – 32
Operating pressure	up to 315 bar
Flow	up to 3500 l/min

Pressure Control Valves

2) Pressure Sequence Valves, Accumulator Charging Valves

Pressure sequence valves, direct and pilot operated

Pressure sequence valves are similar in design to pressure relief valves. The difference is that they are arranged in the main flow, and switch a system on or off when a set pressure is reached.

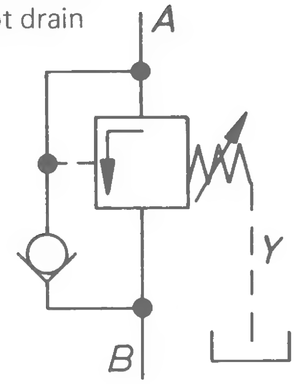
Pressure Sequence Valve, direct operated, type DZ 10 D

The valve comprises a housing 1, control spool 2, pressure spring or springs 3 with adjustment device 4 and check valve 5 (fig. 9).

Compared to the pressure relief valve, the closing element here is a spool, which offers the advantage of fine control.

The spring holds the spool in neutral position and the valve is closed. Pressure in the system at port A is fed via bores and the jet in the spool to the surface opposite the spring.

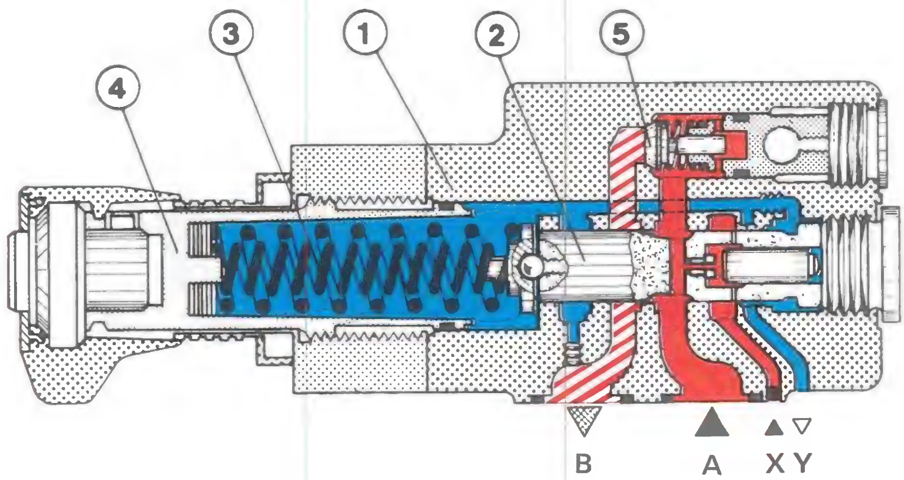
Symbol
Internal pilot supply
External pilot drain



The effective surface is that of the small spool which is supported to the right on the plug screw. If the pressure at A reaches the value set, then the spool moves to the left and opens the connection A to B. The system connected at B is switched on, without a decrease in pressure at A. Use of the small spool in the control spool and thus a small measuring area is related to the pressure rating (spring). At the lowest pressure rating (25 bar max. set pressure), the small spool is not needed, and the large spool area is pressurised. As shown, the small spool is fitted at higher pressure ratings. Two springs are used for 210 bar maximum set pressure.

Supply of fluid can also be external, i.e. from outside via control port X. The jet in the spool must be replaced by a plug. The valve then switches, independently from the inlet pressure, when the pressure at the control port has reached the set value. Depending on the valve application, e.g. as a sequence or shut-off valve, the pilot drain is either external via port Y or internal.

Fig. 9



Pressure Control Valves

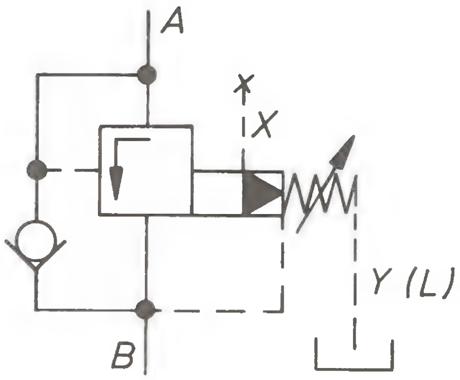
By means of a check valve 5, there can be free flow through the valve from B to A, opposite to the normal switching direction.

Important technical details:
Direct operated pressure sequence valve

sizes	sizes 5, 6 and 10
max. input pressure	315 bar (for size 10)
max. setting range	210 bar
switching pressure	
flow	up to 45 l/min

Pilot Operated Pressure Sequence Valves

Symbol



The pressure valve shown is for use as a sequence valve.

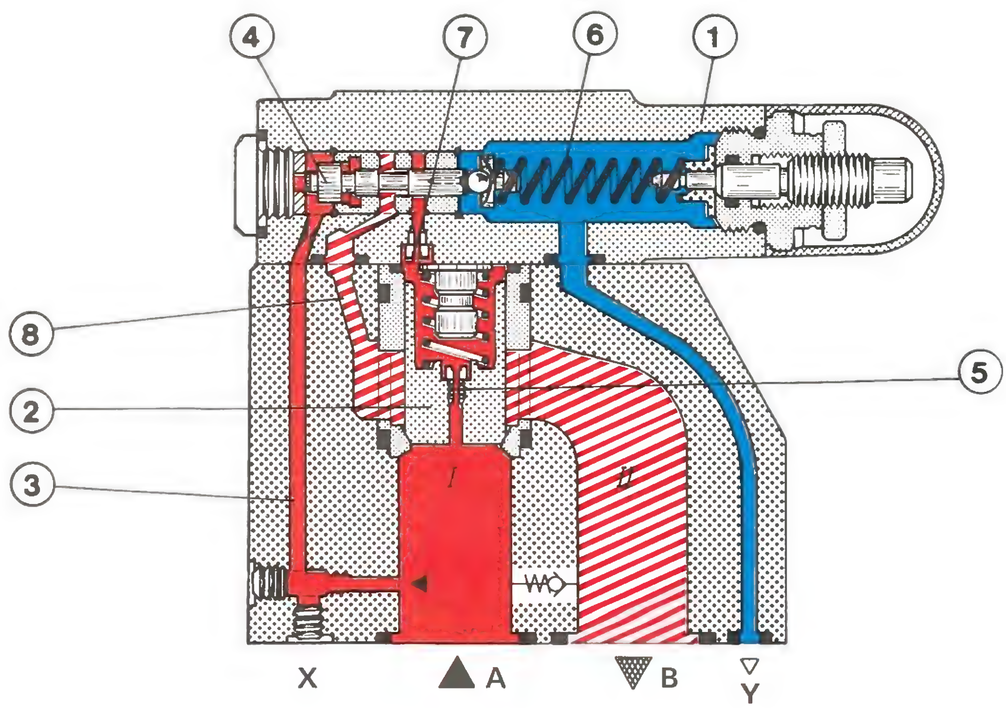
Pilot operated pressure sequence valves are used for larger flows. The pilot valve (1) is a spool valve. (fig. 10)

Pressure from the system (port A) acts on the main spool 2. At the same time, there is also pressure on the spring-loaded side of the main spool by means of a control line 3 at the pilot spool 4 and a bore with jet 5 in the main spool. The spring 6, tensioned according to switching pressure, holds the pilot spool in neutral position. If pressure exceeds the set value, the pilot spool is pushed to the right. Used as a tensioning valve or sequence valve, it allows fluid to drain from the spring chamber into the system II (line B) via jet 7 and control line 8. The jet combination causes a pressure drop between the lower and upper main spool sides. The main spool is pushed upwards. The connection A to B is opened, while system pressure in maintained.

Thus pilot supply is internal in this case, as is pilot drain.

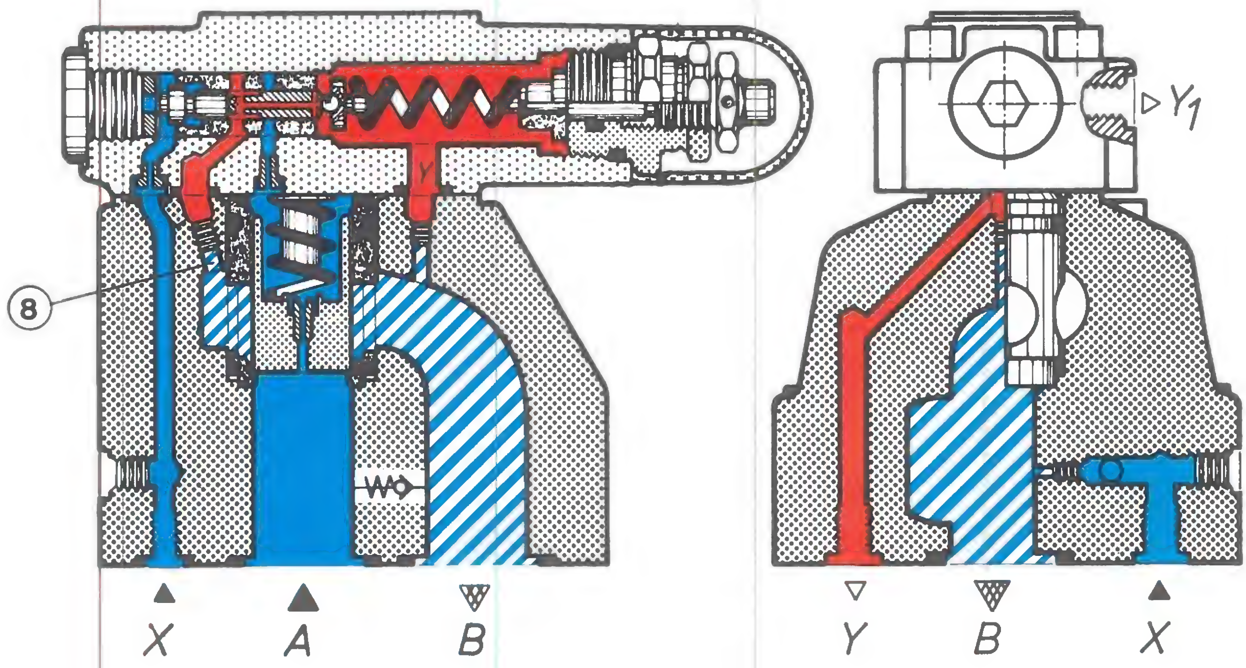
When used as a pre-tensioning valve, leakage oil at the pilot spool is fed internally into line B. When used as a sequence valve, pilot supply is external via port X and pilot drain external via port Y.

Fig. 10



Pressure Control Valves

Fig. 11



Pressure Sequence Valve Type DZ as Bypass Valve

Symbol for this model:

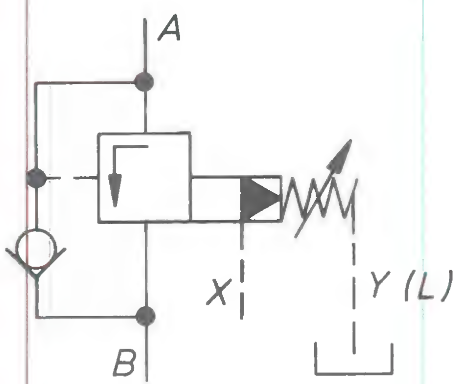


Fig. 11:

Control line 8 is blocked. When the set pressure is reached, the spring loaded side of the main spool is connected with the spring chamber of the pilot valve via bores in the pilot spool. A pressure drop occurs at the main spool, and this lifts from its seat.

A check valve can be fitted optionally for free return flow from line B to line A.

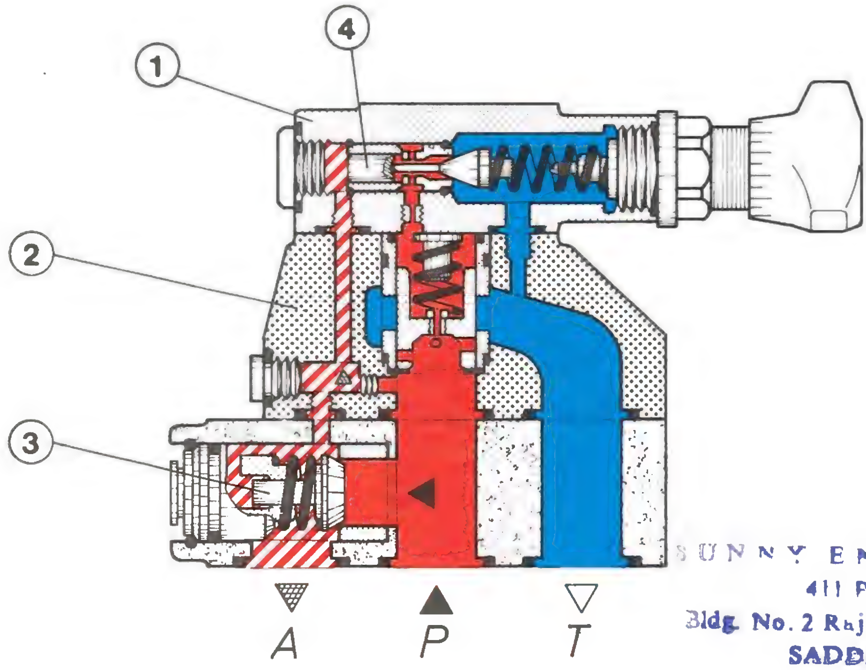
Important technical data:

Sizes	sizes 10, 25 and 32
Max. operating pressure	315 bar
Max. set switching pressure	210 bar
Flow	up to 450 l/min

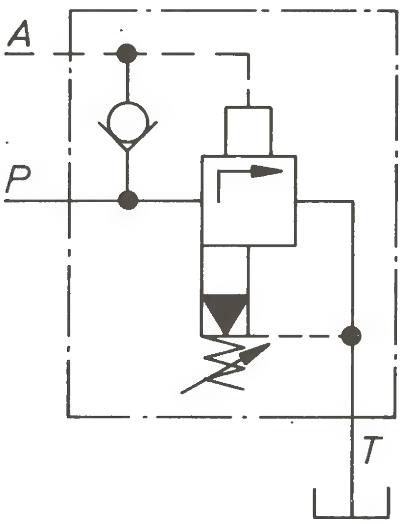
Pressure Control Valves

Pilot Operated Accumulator Charging Valves, Type DA (Pressure Shut-Off Valves)

Fig. 12



Symbol



Pressure shut-off valves type DA are used in hydraulic systems with pressure accumulators. They comprise pilot valve 1, main valve 2 and check valve 3 (fig. 12).

They serve to feed pump flow into the accumulator circuit until the accumulator is filled and the necessary accumulator pressure is reached.

Oil now flows from P via the check valve to the accumulator connected at A. The accumulator pressure (shut-off pressure) is set at the pilot valve. Pressure acts on the spring loaded rear side of the main spool via jet in the main spool, also via a further jet at the pilot valve. If the pilot poppet rises from its seat when switching pressure is reached, a pressure drop occurs at the main spool when the control oil flows.

This moves upwards and allows the flow to drain to tank. The accumulator circuit is blocked by the check valve. In order that the pilot poppet, now unloaded, does not close again, the pressure in the accumulator circuit acts on the small spool 4 in the pilot valve via the control line behind the check valve (red). The pilot poppet remains pushed back via the spool 4 and plunger, until accumulator pressure has decreased due to fluid drain according to the difference in area between spool 4 and the pilot poppet. The main poppet then closes again and the accumulator is re-filled.

Important technical data:

Sizes	sizes 10, 20, and 30
Operating pressure	up to 315 bar
Flow	up to 250 l/min

Pressure Control Valves

3) Pressure Reducing Valves

These valves are also known as "pressure regulating valves"

Pressure reducing valves, direct and pilot operated

A pressure reducing valve is used to limit secondary pressure, i.e. the output pressure. The secondary pressure is held constant independent of input pressure (primary pressure) as soon as this reaches the value set. It is therefore possible to reduce pressure in one part of the system to a value lower than system pressure.

Pressure reducing valves, direct operated (type DR ... D)

The control element is a spool 1, which is held in neutral position in the housing 2 by the spring 3.

Contrary to the pressure relief and pressure sequence valves, the pressure reducing valve is open in neutral position (fig. 13).

Oil flows from P to A. The secondary pressure (port A) affects the left spool surface via control line 4. If pressure at A reaches the value set at the spring, the spool moves and reduces the flow from P to A. Oil can be taken from A by the user for any pressure in the user line up to the set pressure. If the user does not take any fluid, e.g. at end position, the valve is closed. If we are dealing with a 3 directional model pressure reducing valve, as shown here in the sectional diagram, there is simultaneous pressure protection of the secondary circuit. If pressure behind the valve increases due to the effect of external forces on the user, it is pushes the control spool further against the spring. Port A is thus connected to tank. Fluid drains until the set pressure is once again established.

Symbol

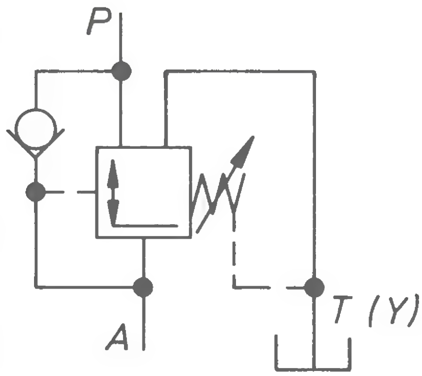
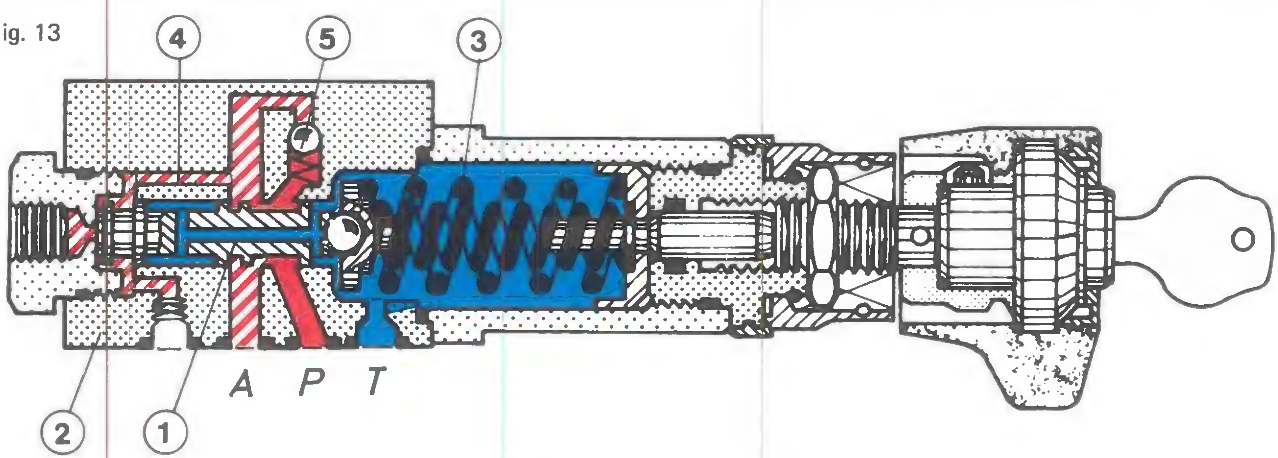


Fig. 13



The sectional diagram shows a pressure reducing valve type DR 6 D

Pressure Control Valves

A check valve 5 is fitted parallel to the user, for independent return of fluid from the user.

Important technical data:

Sizes	sizes 5, 6 and 10
Max. input pressure	315 bar
Max. output pressure	210 bar
Flow	up to 45 l/min

Pilot Operated Pressure Reducing Valve (type DR)

Pilot operated pressure reducing valves are used for reducing pressure with large flows. The pilot valve 1 is a direct operated pressure relief valve. The main valve has a spool 2, which guarantees free flow from B to A in neutral position (fig. 14).

The required output pressure is set at the spring 3 of the pilot valve. Pressure at A acts on the underside of the spool. It also affects the pilot poppet 7 and the spring loaded upper side of the main spool via the control line 4 with the jets 5 and 6. As long as the input pressure is less than the set pressure, the main spool remains open by means of spring 8. If pressure at A reaches the set value, the pilot valve opens, and pilot oil drains. The pressure drop which occurs allows the main spool to travel upwards, progressively closing the valve. Only a certain amount of oil flows to the outlet, so that the pressure at A is not exceeded. If the user does not take any fluid, the main spool is closed. Pilot oil flows continuously to tank via the pilot valve during the regulating function.

The check valve has free flow in direction A to B.

Important technical details:

Sizes	sizes 8 to 32
Max. set pressure	315 bar
Flow	up to 300 l/min

Symbol

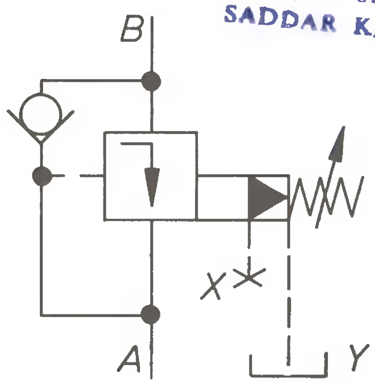
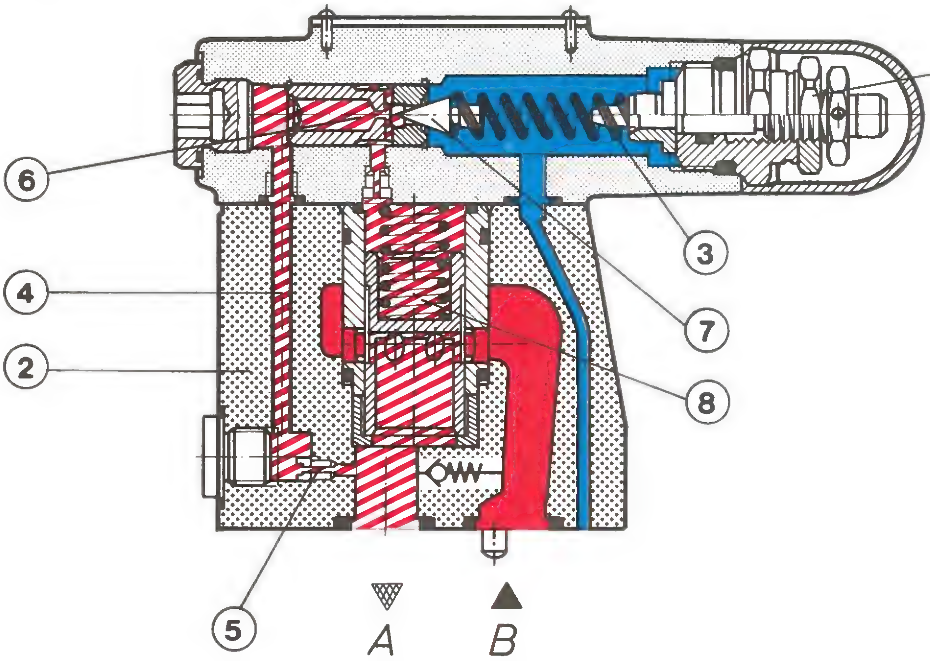


Fig. 14

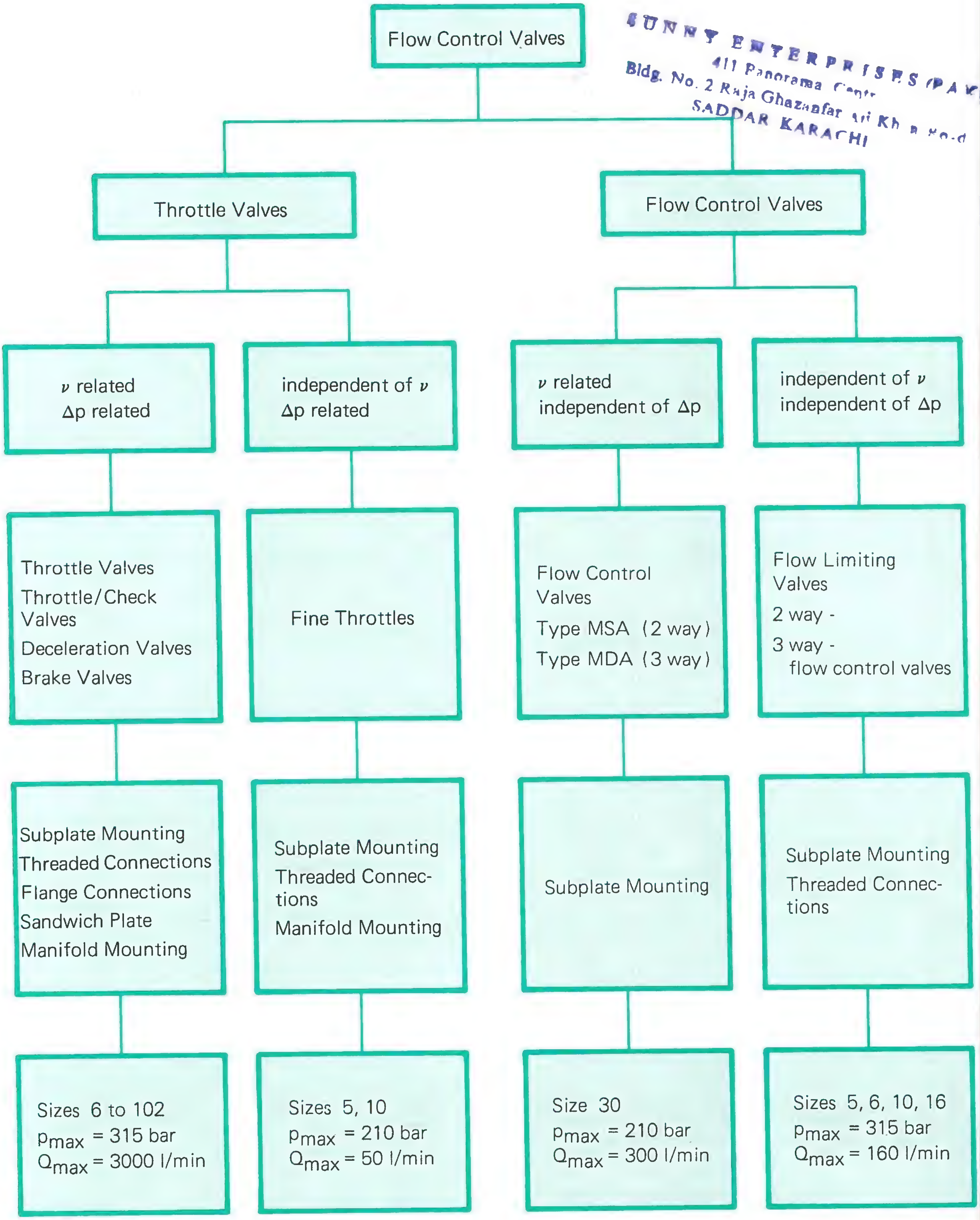


Pilot operated pressure reducing valve DR 10

Notes

Notes

Programme Summary



Flow Control Valves

Flow control valves serve to influence the speed of movement of users by restricting the oil flow.

Stepless speed control is thus possible.

Flow control valves can be divided into 4 groups:

1) pressure and viscosity related
2) pressure related, but independent of viscosity

} throttle valves

3) independent of pressure, but viscosity related
4) independent of pressure and viscosity

} flow control valves

Throttle Valves

While the flow restriction remains the same, the flow changes according to the pressure drop at the throttle position.

Flow Control Valves

While the flow restriction remains the same, the flow remains constant, independent of the pressure drop at the flow control valve.

Flow at the Throttle Positions

The flow at a throttle position is calculated according to the equation:

$$\boxed{Q = \alpha \cdot A \cdot \sqrt{\frac{\Delta p \cdot 2}{\rho}}}$$

$$\alpha = \sqrt{\frac{1}{\zeta}}$$

$$\zeta = \frac{l \cdot 64 \cdot \nu}{v \cdot d_H{}^2}$$

(for laminar flow)

Q = flow
A = throttle sectional area
Δp = pressure loss (pressure drop between A and B)
α = flow coefficient
ρ = density
ζ = resistance coefficient
l = throttle range
ν = viscosity
v = flow speed
d_H = hydraulic diameter
$= 4 \frac{A \text{ (throttle section)}}{U \text{ (flow path)}}$

According to the type of throttle, a flow coefficient of 0.6 – 0.9 can be used for jets and orifices.

Throttle Valves

The flow of throttle valves is related to the pressure drop at the throttle position, i.e. a larger Δp results in a larger flow. The equation for the resistance coefficient shows the relationship to the viscosity. The shorter the throttle length l, the less noticeable is a change in viscosity. It should also be noted that the flow increases as the fluid becomes thinner.

Whether a valve is dependent on the viscosity or is practically independent, depends on the construction of the throttle position.

Throttle valves are used, when:

– there is constant working resistance
– change in speed is irrelevant or desired with changing load.

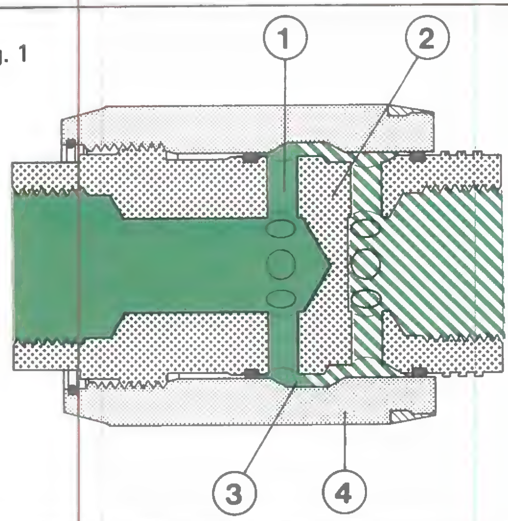
Flow Control Valves

Simple Throttle, type MG and Throttle/Check Valve, type MK

Valves type MG and MK for line mounting; in various sizes (size 6 – 30)

Valve type MG for sizes 52 – 102

Fig. 1



Simple throttle, type MG

Symbol

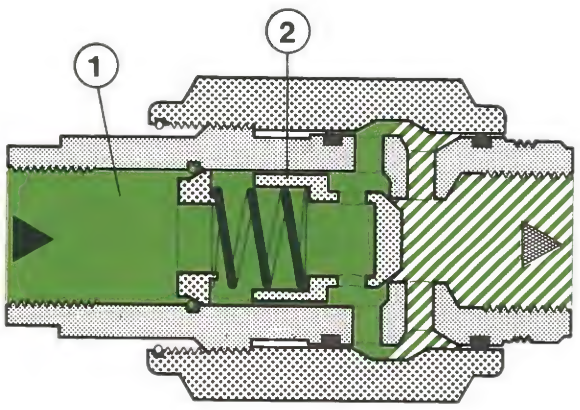
These throttle valves are related to pressure and viscosity.

Oil reaches the throttle position 3 by means of side bores 1 in the housing 2. These are formed between the housing and the adjustable sleeve 4. By turning the sleeve, the ring-shaped section at the throttle position can be altered steplessly. There is throttling in both directions (fig. 1).

Flow Control Valves

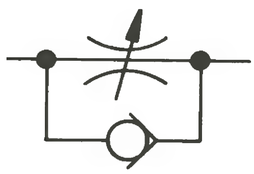
If throttling is required in one direction only, an additional check valve is necessary.

Fig. 2



Throttle/Check Valve Type MK

Symbol



In throttle direction, fluid reaches the rear side 1 of the valve poppet 2. The poppet of the check valve is pushed on its seat. Throttling procedure is as per type MG (fig. 2).

In the opposite direction (right to left) the flow acts on the face surface of the check valve. The poppet is lifted from its seat. Oil flows unthrottled through the valve. At the same time, part of the fluid passes over the ring slot and thus the desired self-cleaning process is achieved.

Important technical data:

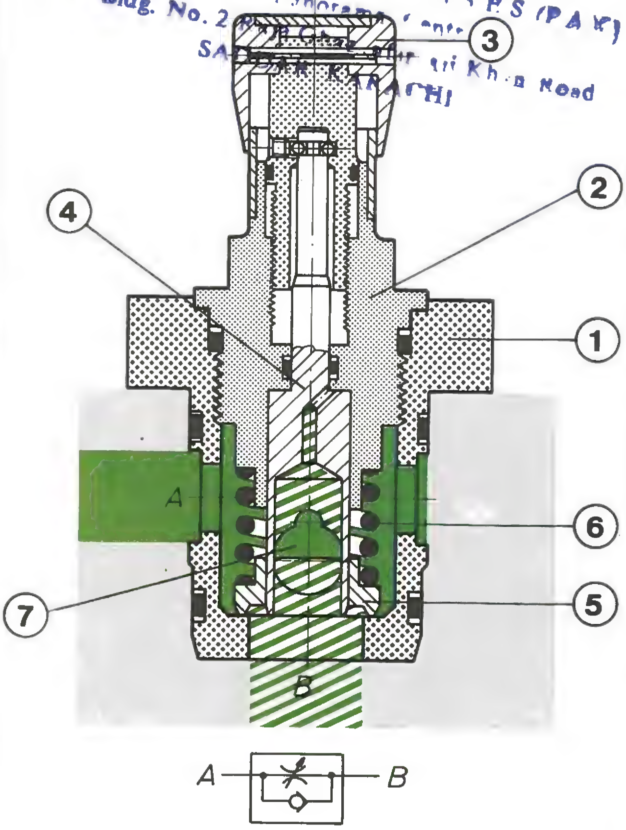
Sizes	sizes 6 to 102
Flow	up to 3000 l/min
Operating pressure	up to 315 bar

Throttle and Throttle/Check Valve for Manifold Mounting

This valve is fitted in a manifold insert and therefore does not have its own housing. The valve is fitted in a mounting hole (fig. 3).

The photograph shows the throttle / check valve model.

Fig. 3



Throttle/Check Valve

The throttle/check valve comprises a cartridge bush 1, valve body 2 with adjustment head 3 and throttle pin 4, also check valve 5 with spring 6.

The throttle direction is from A to B. The throttle section is formed by the throttle pin with circular

recess 7 and the check valve ring 5. When the adjustment knob is turned, the throttle pin moves vertically and alters the throttle section.

The throttle range is relatively short, so that the viscosity influence is not very great.

When there is flow from B to A the check valve ring is pushed upwards. Oil flows without throttling to port A.

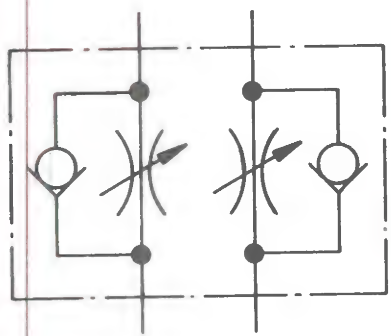
Important technical data:

Sizes	sizes 10, 20, 30
Flow	up to 400 l/min
Operating pressure	up to 315 bar

Double Throttle/Check Valve type Z 2 FS

Double throttle/check valves comprise two throttle/check valves arranged symmetrically in one block.

Symbol



They are fitted between the direct operated directional valve and the subplate to influence the speed of a user (main flow limiter).

With pilot operated valves, the double throttle/check valve can be used as pilot choke adjustment (pilot flow limiter). It is then fitted between pilot and main valve. (See also the sectional diagram for pilot operated directional valves.)

With flow from the bottom to the top, pressure acts via bore 1 on the mounting face of the check valve, designed as a throttle pin. The throttle pin is pushed back and no throttling takes place (fig. 4).

With flow from the top to the bottom pressure acts via bore 2 on the rear side of the throttle pin. It is pushed against the stop 3 and occupies a throttle position according to the position of adjustment screw 4.

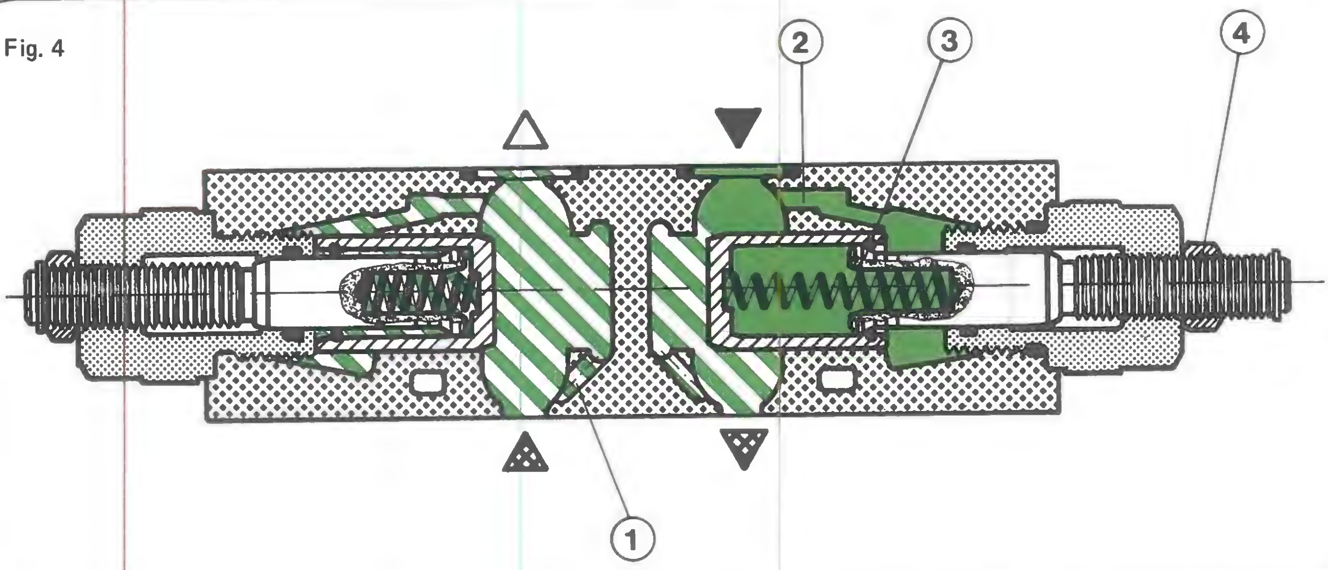
Throttling can take place in the supply or drain, depending on the position of the sandwich plate (size 5 and 10) or arrangement of the throttle cartridges (size 16 and 22).

Important technical data:

Sizes	sizes 6, 10, 16, 22
Flow	up to 300 l/min
Operating pressure	up to 315 bar

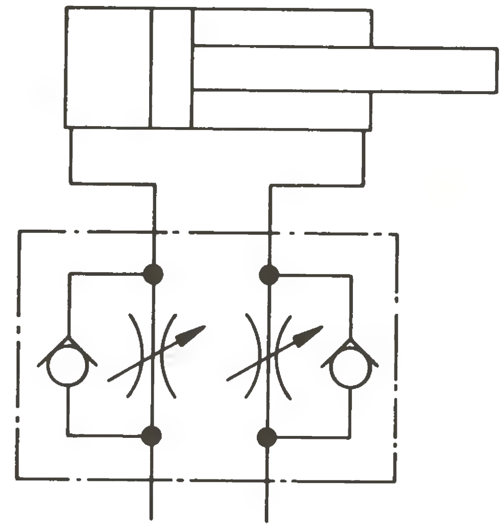
Double Throttle/Check Valve Type Z 2 FS 22

Fig. 4



Flow Control Valves

Fig. 5



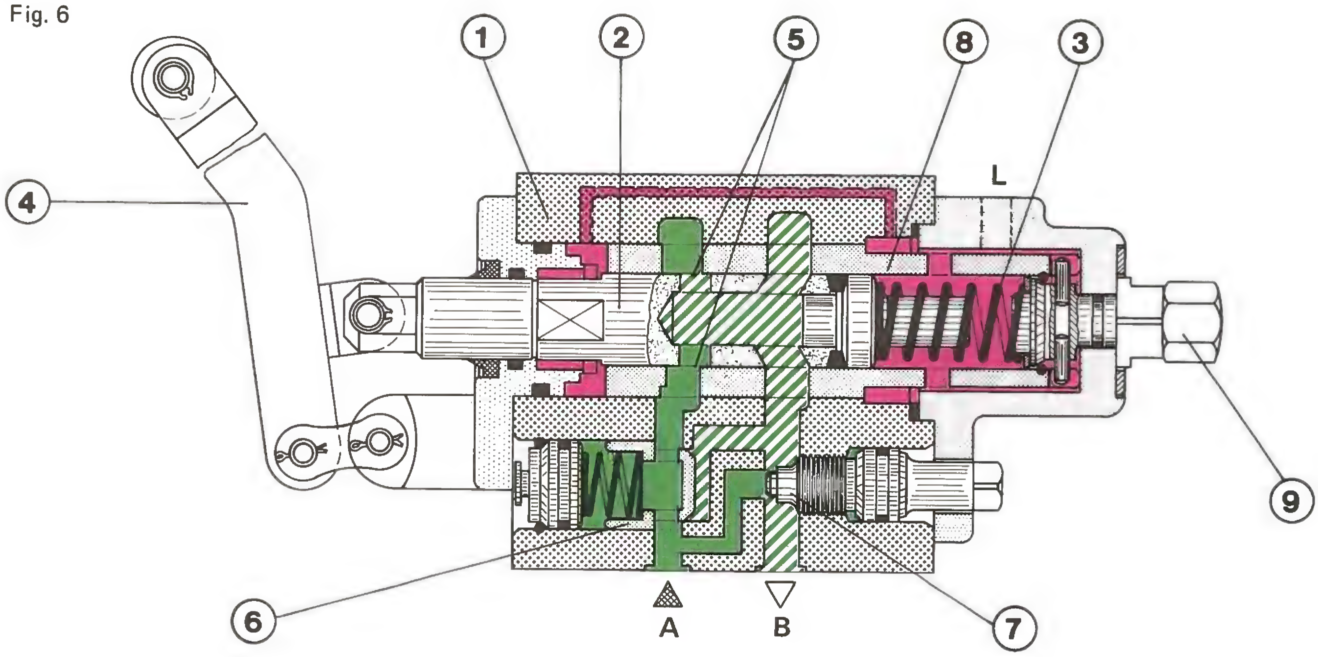
Meter-in control

Meter-out control

Deceleration valves, type FM

Deceleration valves, with roller operation, type FMR (left)
and with roller/lever operation, type FMH (right)

Fig. 6

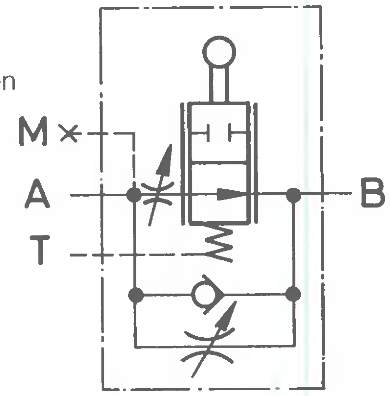


Flow Control Valves

Deceleration valves serve to decelerate or accelerate steplessly hydraulically moved weights independent of direction.

The following model is shown:
neutral position open
with check valve
with main flow throttle
with secondary flow throttle

Symbol



The throttle spool 2 is pushed to the left into its neutral position by the spring 3 in a housing 1. (fig. 6)

According to the spool type, the connection A to B is open in neutral position, as shown, or closed.

Basic function:

The cylinder, whose speed is to be influenced, operates the roller/lever 4 of the deceleration valve by means of a cam on the piston rod.

Fig. 7 shows the arrangement.

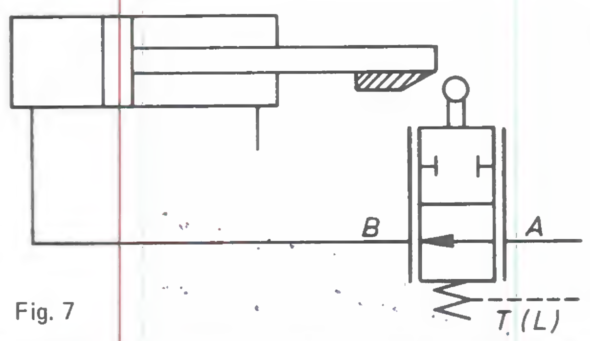


Fig. 7

The throttle spool is pushed back against the spring. Flow restriction area 5 therefore decreases as the piston stroke increases. The cylinder speed decreases.

If the connection A to B is completely closed, the cylinder remains still. It has interrupted the oil supply.

The deceleration is related to the type of cam at the piston rod.

In order to allow the cylinder to travel out of closed position, a check valve 6 can be arranged parallel to the throttle spool. It guarantees free flow from B to A. The cylinder then travels at maximum speed out of end position.

If the throttle section is not quite closed, the end position of the cylinder should be limited mechanically. If, in such a case, a check valve has not been provided, there is controlled acceleration when travelling out of end position.

The deceleration valve can also be fitted optionally with a secondary flow throttle 7 and/or with a main flow throttle 8.

Model with Secondary Flow Throttle:

Where secondary flow throttle 7 us used, a smaller secondary flow is set with throttle spool 2 closed. After the main flow has been closed, the cylinder can travel at "creep" into the mechanically limited end position.

Model with Main Flow Throttle:

The main throttle 8 is a sleeve with radial bores. When the adjustment screw 9 is turned, the sleeve bores move opposite to the bores in spool 2. The maximum flow section can thus be changed and the valve adapted to suit the oil flow. This causes the complete deceleration stroke to be available, even with a low oil flow.

As the deceleration valve is a spool valve, leakage occurs on both sides of the spool. The leakage oil must be drained to tank via port L.

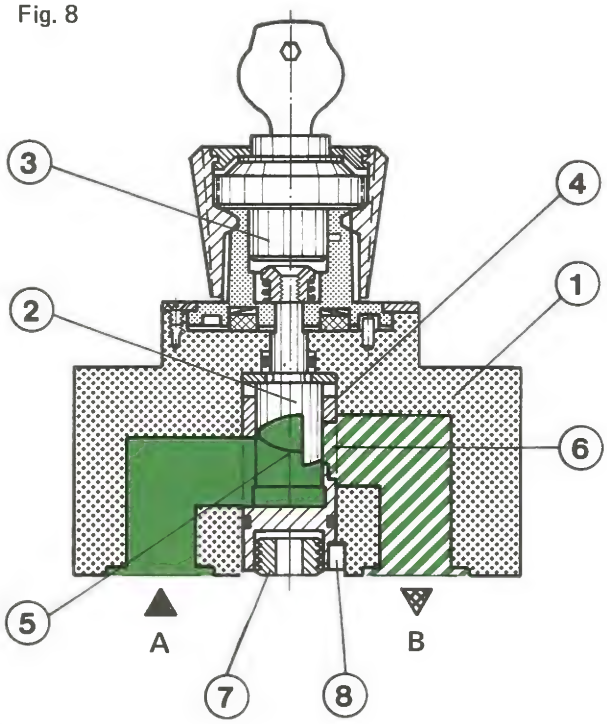
Flow Control Valves

Fine Throttle Type F

Fine throttle as cartridge unit

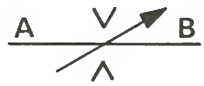
The fine throttle belongs among flow control valves in group 3 — flow is related to pressure and almost independent of viscosity. The viscosity influence is very slight, due to the orifice type throttle position.

Fig. 8



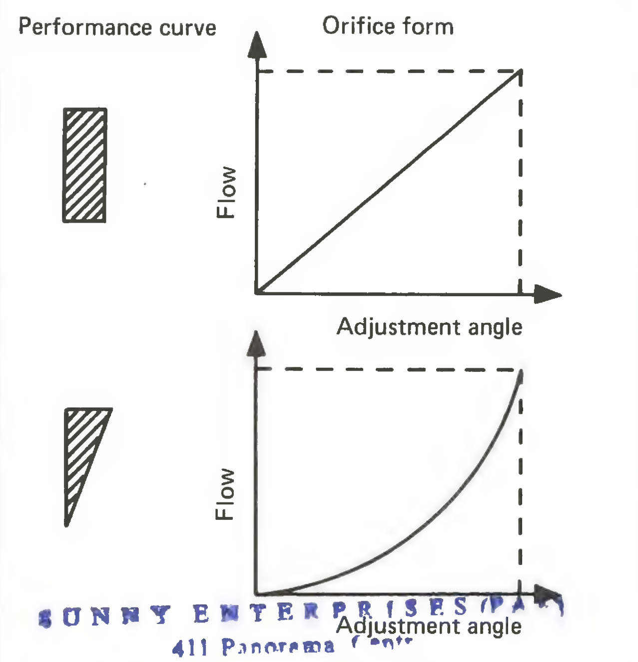
The sectional diagram (fig. 8) shows a valve with housing. The curved pin 2, setting element 3 with scale and orifice 4 are fitted in the housing.

Symbol



The throttle is adjusted by turning the curved pin, which is joined to the adjustment knob. The flow section is determined by the position of the curved pin, i.e. by the position of curve 5 in front of the orifice window 6.

There is a linear or non-linear flow curve over the adjustment angle (300°) according to the orifice type.



The direction of flow is preferably from A to B. By means of an adjustment screw 7, the orifice opposite the curved pin can be raised or lowered. Adjustment of the setting device is therfore guaranteed. During operation, the orifice with adjustment screw is supported on the valve mounting face.
A pin 8 is fitted to ensure that the orifice cannot turn in the wrong direction.

Important technical data:

Sizes	sizes 5 and 10
Flow	up to 50 l/min
Operating pressure	up to 210 bar

Flow Control Valves

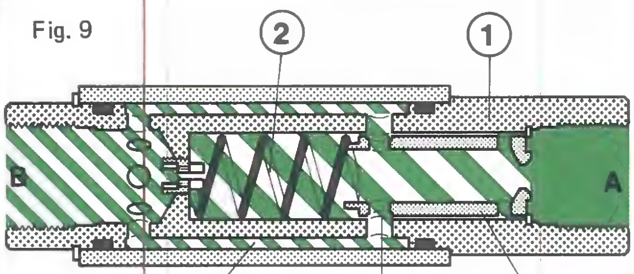
Flow regulating valves

In flow regulating valves, the flow is not related to the pressure drop between the valve input and output. This means that the oil flow set remains constant, even with pressure deviations.

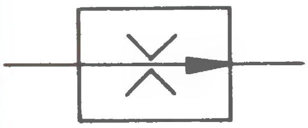
Flow regulating valves are therefore used when the working speed should remain fixed in spite of different loads at the user.

2 Directional Flow Limiting Valve, type 2 FB ... F

Fig. 9



Symbol



The valve comprises a housing 1, spring 2 and orifice bush 3. The oil flows from the orifice side A into the valve then via side bores 4 and an annulus port 5 on to the valve outlet (fig. 9).

The flow restriction is fixed by the selection of the orifice. A pressure drop occurs at the orifice during flow. The orifice 3 is pushed against the spring.

As flow increases, i.e. increase of Δp, the flow area of the side bores 4 is decreased in size according to the increase in pressure drop. The flow thus remains fixed.

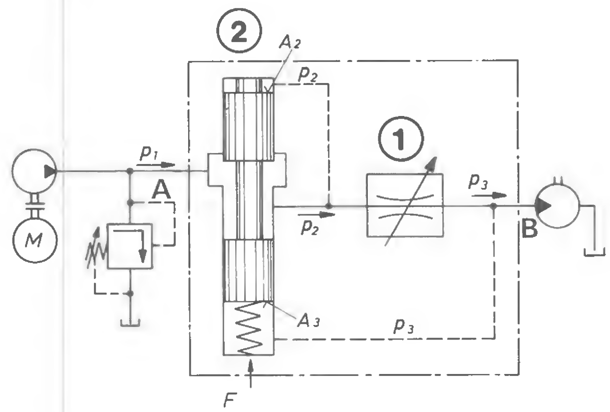
A variable model of this valve is also available (type 2FB-V). By increasing the spring tension (adjustable) and thus increasing the pressure drop necessary to move the orifice, it is possible to change the flow by approximately 25%.

2 Directional Flow Regulating Valves, type 2 FRM

2 Directional flow regulating valves sizes 5 and 10, operation by means of roller/lever and rotary knob.

In order to understand the function better, it will firstly be described using a circuit diagram.

Fig. 10



Functional diagram
2 directional flow regulating valve

In this case, a regulating spool 2 is controlled before the actual adjustable throttle 1 (fig. 10). Flow is from A to B.

In each case, the maximum system pressure p_1 builds up in front of the valve. Pressure p_3 is behind the valve, according to the working resistance at the user. When the working resistance changes, pressure p_3 also changes and thus also the pressure drop from p_1 to p_3.

Flow Control Valves

While the throttle section remains the same, different volumes would therefore flow, if there were no regulating spool.

It must be ensured that there is always the same pressure difference $p_2 - p_3$ at the throttle position, in order to prevent the influence of pressure deviations.

This is achieved, using the control spool, also called pressure compensator, as an adjustable throttle element.

A spring pushes the spool in opening direction and holds it in neutral position where there is no flow through the valve. If there is flow through the valve, the pressures exert a force on the spool via surfaces A_2 and A_3.

Pressure p_2 in front of the throttle position affects the surface A_2 by means of a control line.
The pressure behind the throttle position affects the surface A_3 by means of a control line.

The following forces result at the control spool:

in opening direction (upwards) $F_{spring} + p_3 \cdot A_3$

in closing direction (downwards) $p_2 \cdot A_2$

In control position, i.e. when there is flow through the valve, the forces at the spool are balanced.

Conditions for equilibrium:

$$p_2 \cdot A_2 = p_3 \cdot A_3 + F_{spring} \; / : A$$

$$A_2 = A_3 = A$$

$$\rightarrow p_2 = p_3 + \frac{F_{spring}}{A}$$

$$\rightarrow p_2 - p_3 = \frac{F_{spring}}{A}$$

The conditions for equilibrium show that the pressure drop $\Delta p = p_2 - p_3$ always results at the throttle position corresponding to the spring force.

As only slight spring deflections occur, the spring force can be assumed to be almost constant.

If pressure p_2 increases, for example due to an increase in pressure at the valve inlet, the regulating spool moves in closing direction. Thus it decreases the quantity of oil flowing to the throttle position until pressure p_2 has fallen again and the pressure drop $p_2 - p_3$ corresponds to spring force : surface area A.
The flow therefore remains constant.

If, for example, pressure p_3 changes (increases or decreases) the spring again moves until the pressure drop $p_2 - p_3 = \frac{F_{spring}}{A}$ has been reached again.

The excess fluid in front of the valve must be drained by means of the pressure relief valve, as with the throttle described previously.

Fig. 11 shows the actual valve.

Fig. 11a

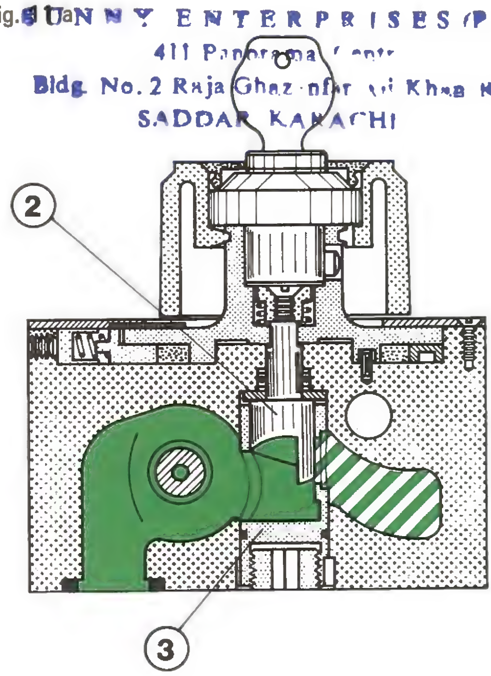
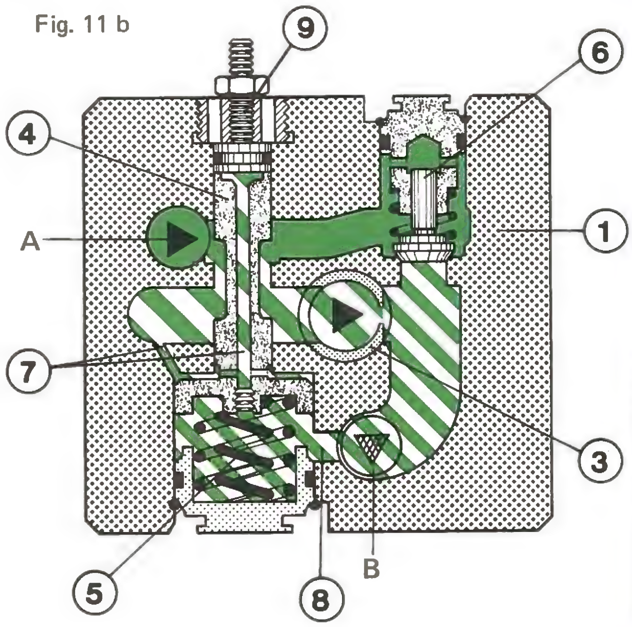
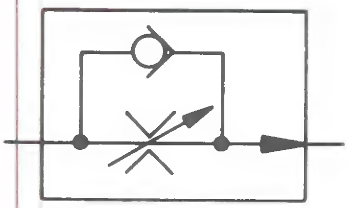


Fig. 11 b



Flow Control Valves

Symbol



The throttle pin 2 with orifice 3 (as for the fine throttle) and the control spool 4 with spring 5 are fitted in a housing 1.

An additional check valve 6 is fitted, to guarantee free flow from B to A. The regulating spool is effective only in flow direction A to B.

Pressure in front of the throttle position (p_2) is fed by means of bores 7 to the spool surfaces, which lie opposite the spring.

Pressure behind the throttle position (p_3) affects the spool by means of bore 8, in addition to the spring.

To avoid surges, i.e. the regulating spool springs out of neutral position beyond the actual control position, if there is sudden flow through the valve, the spool can be fixed at standstill in a set regulating position by means of a stroke limiter 9.

Important technical data:

Sizes	sizes 5, 10, 16 (also over size 30 with ν-dependent throttle position)
Flow	up to 160 l/min
Operating pressure	up to 315 bar

3 Directional Flow Control Valve

The basic function corresponds to that of the 2 directional flow control valve.
However, compared to the 2 directional flow regulating valve, the 3 directional valve has an additional tank port T.

Symbol simplified

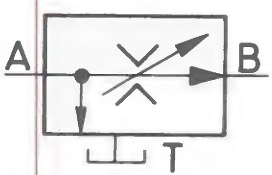
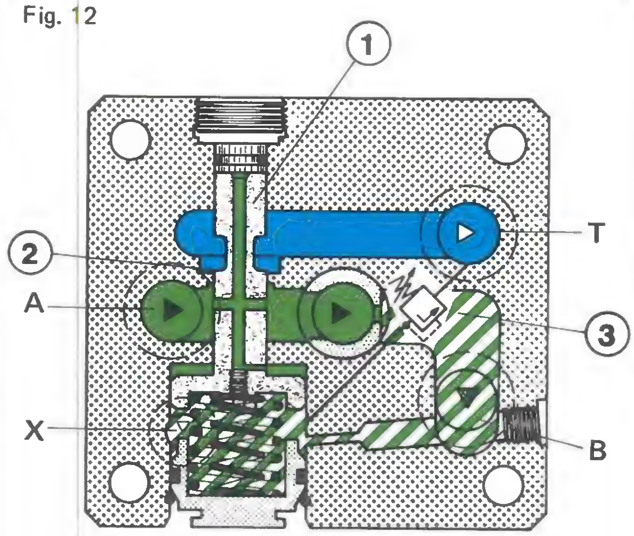


Fig. 12



The regulating spool 1 is arranged in such a way that the pump flow which is not required is drained direct to tank by means of control lands 2. The pump must only act against the load pressure + pressure drop, which increases the degree of efficiency.

Three directional flow control valves can be used only in the supply line or pilot line to a user.

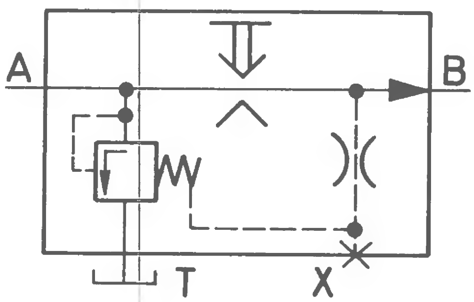
An unloading port X for almost free flow is possible on this valve version.

The sectional diagram shows the model with built-in pressure relief valve 3 (overload protection).

Important technical data:

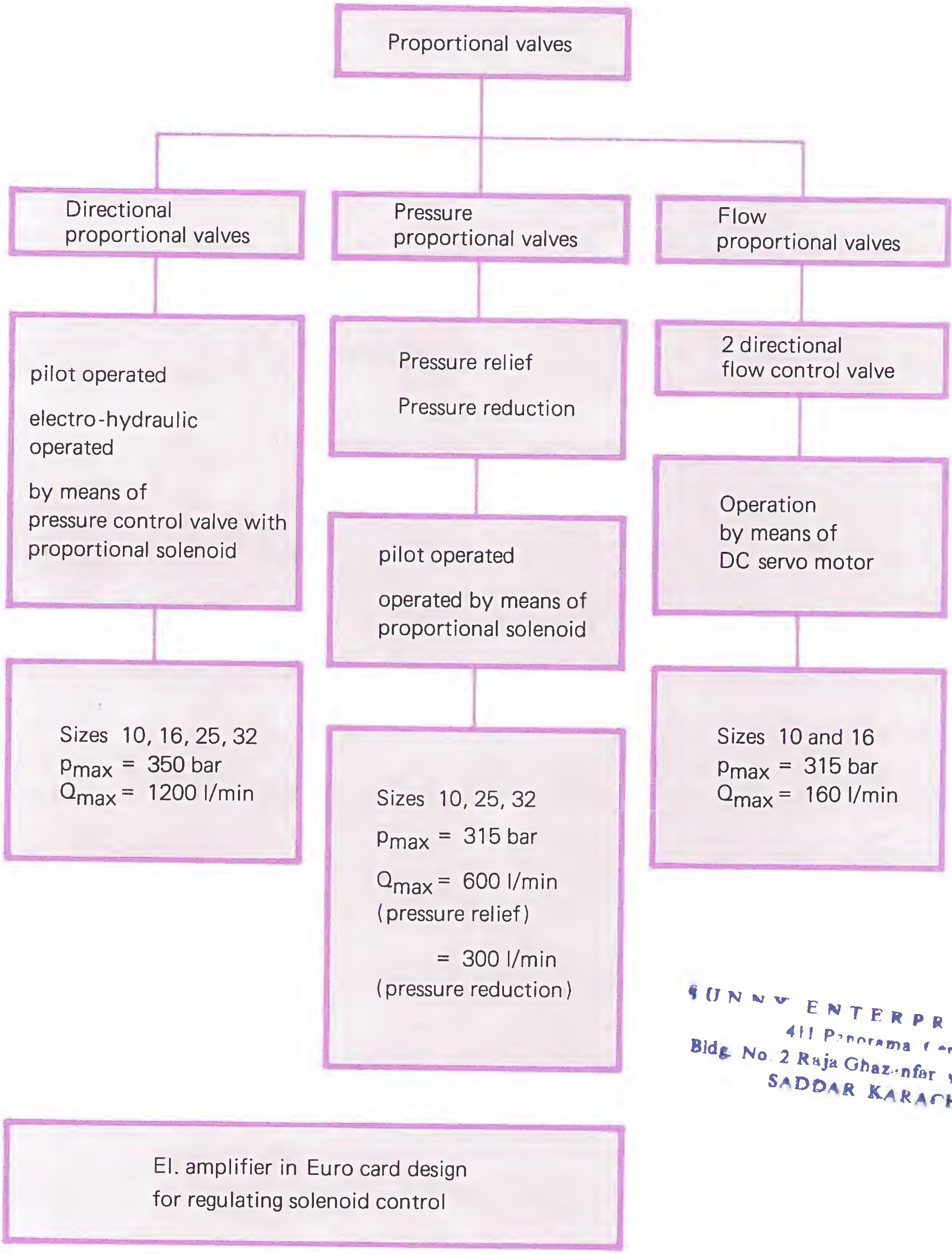
Sizes	size 10 and 16
Flow	up to 160 l/min
Operating pressure	up to 315 bar

detailed

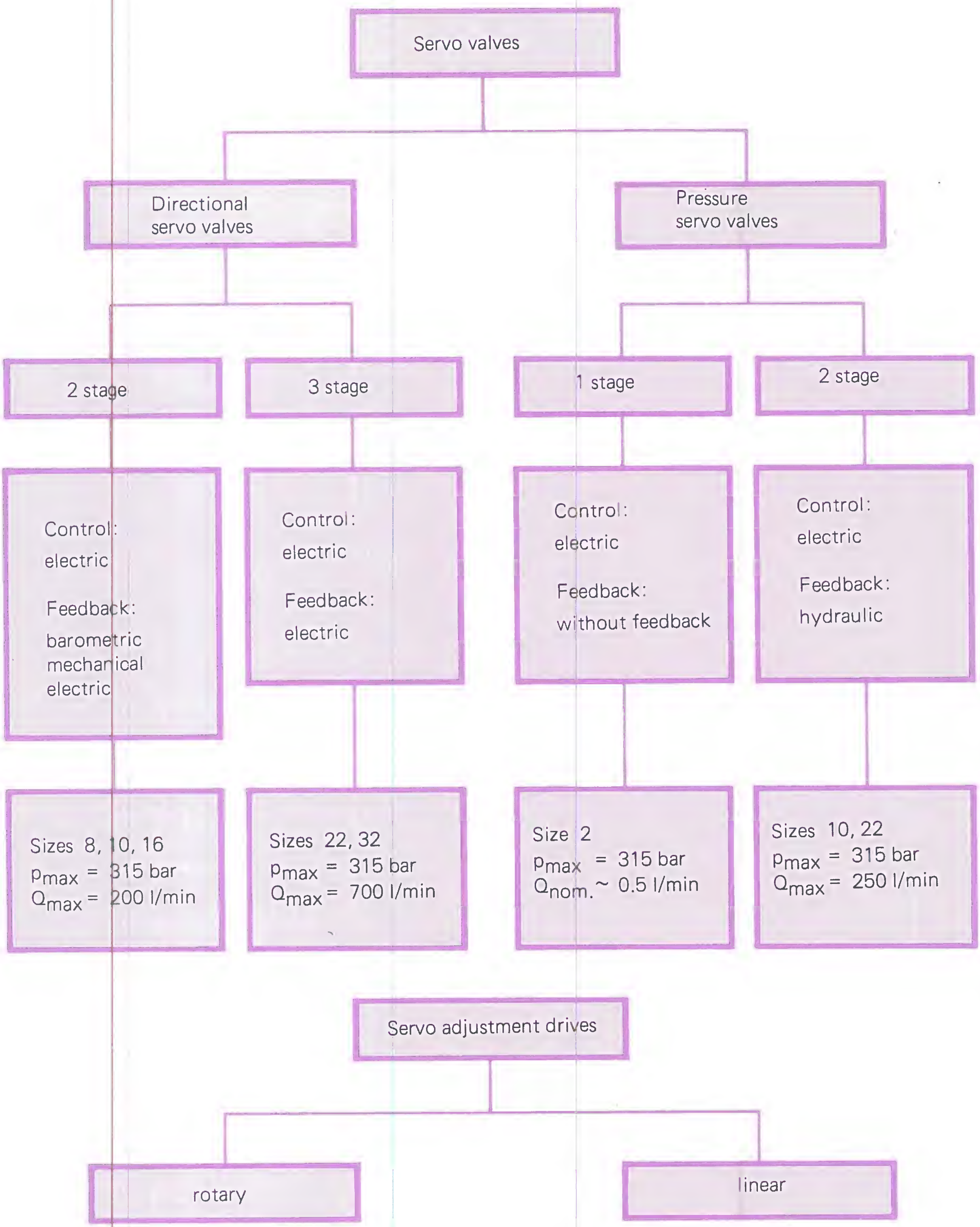


Notes

Programme Summary

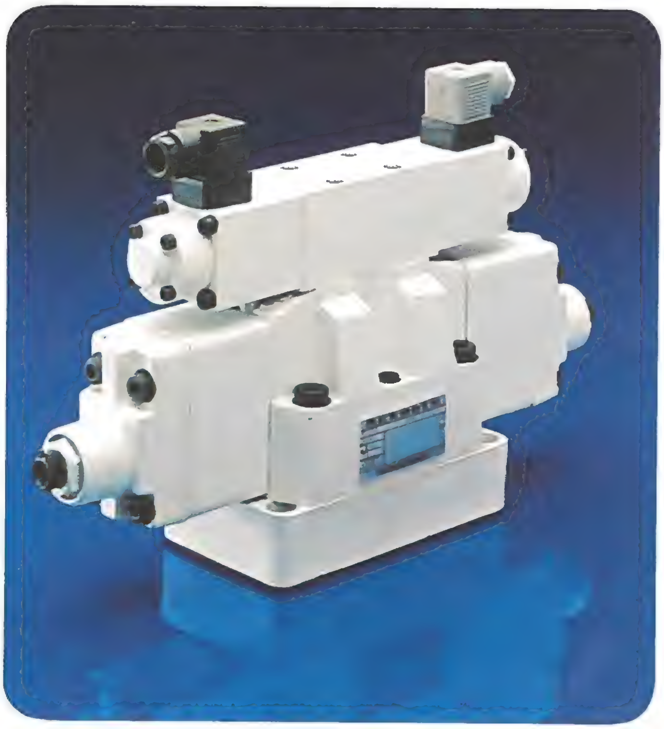


Programme Summary



Proportional Valves

Directional Proportional Valves

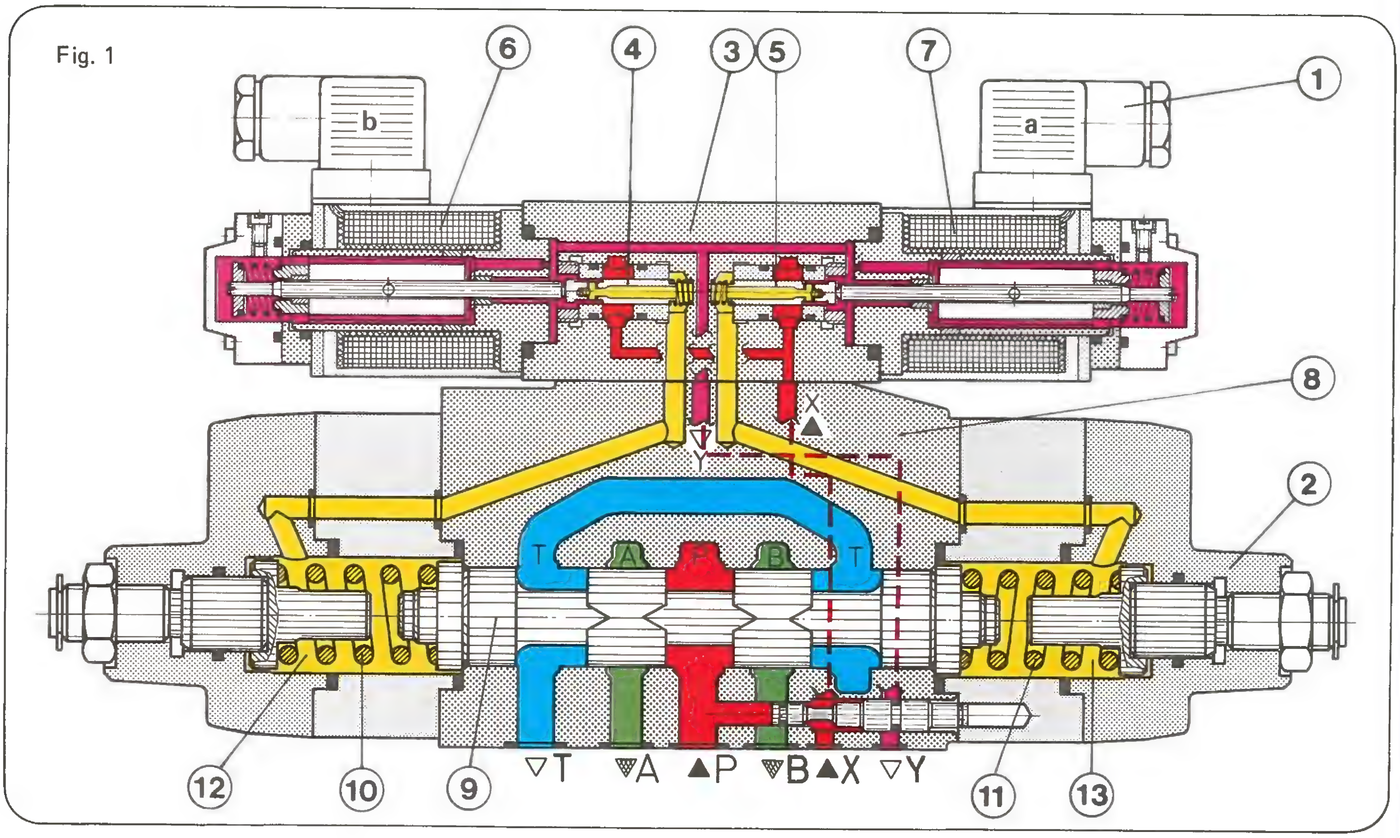


4 directional proportional valve, type 4 WRZ 16.M..

Complex control produces and programmed sequences of a user, such as acceleration — movement — deceleration, can be controlled using a directional proportional valve. One unit determines the direction of movement and the speed.

The output flow is proportional to the electrical input signal. Servo units behave the same and, as with servo valves, proportional valves not only control the operation, but also the regulation of a hydraulic circuit.

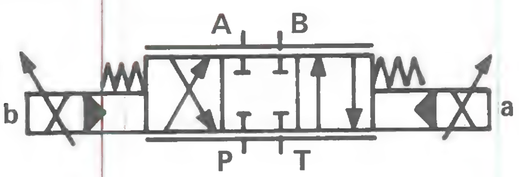
Fig. 1



Proportional Valves

The 4 directional proportional valve consists of a pilot valve 1 and the main valve 2 (fig. 1).

Symbol
X = internal
Y = external



The pilot valve is a pressure regulating valve with proportional solenoid, i.e. an adjustable oil-immersed DC solenoid.

It changes an electrical input signal into a proportional force. An increase of current causes a corresponding increase in solenoid force.

In order to achieve smooth power delivery, only a part of the solenoid stroke available is used.

The pilot valve comprises a housing 3, two pilot spools 4 and 5 and two proportional solenoids 6 and 7.

The main valve is a directional control valve and comprises the housing 8, main spool 9 and centering springs 10 and 11.

At input signal 0, both spring chambers of the main valve 12 and 13 are unloaded to the Y port via bores in the pilot spool.

The main spool 9 is held in centre position by the centering springs 10 and 11.

If, for example, solenoid 6 is energised, the pilot spool 4 is pushed to the right. Pilot oil (internal from line P or external by means of port X) now reaches the spring chamber 4 by means of the cross and horizontal bore in the pilot spool. At the same time, the pilot spool closes the oil path from the main spring chamber to port Y.

Pressure builds up in the spring chamber according to the solenoid force (see pressure regulating valve, only that here the spring force is replaced by solenoid force).

The pilot pressure, proportional to the input current, pushes the main spool 9 against the spring 11, until the pressure and spring force are equal.
Higher solenoid force → higher control pressure → longer spool stroke.

A certain flow therefore results, related to the input current.

Orifice type throttle sections with non-linear flow characteristics result from the spool type.

Special mention must be made of the switching and transition behaviour. The change-over from the closed to the open position is always controlled. There are no step type transitions during the opening phases as with standard directional control valves. The metering notches and spool control are arranged in such a way that controlled opening and closing of the flow porting occurs in a progressive manner.

The return of pilot spool 4 to neutral position at signal 0 and thus also the return of the main spool to its neutral position are not related to the control pressure.

The directional proportional valves can also be fitted with a pressure compensator. It is arranged in a sandwich plate under the proportional valve. In this way, a flow characteristic corresponding to that of a flow control valve is achieved, which is not related to the pressure drop at the throttle position.

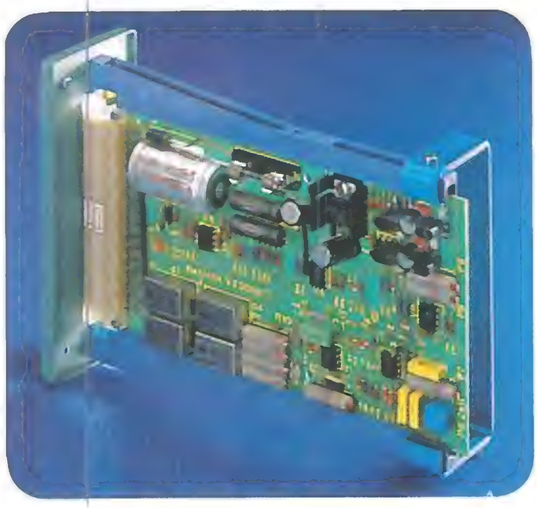
The pressure compensator can be used in the supply line as a 2 or 3 directional flow regulator, or in the return line as a 2 directional flow regulator (for the pressure compensator principle, see flow control valves)

A pressure compensator can also be used for several proportional valves built on a sandwich plate.

Important technical data:

Sizes	10 – 32
Operating pressure	up to 350 bar
Flow	up to 1200 l/min

Directional proportional valves are generally controled by means of the electrical amplifier type VT. 3000.



Proportional Valves

It comprises: (fig. 2)

1 Supply voltage, voltage controller

2 Call-up relay d1 – d4 to switch over the nominal values

3 Internal nominal value trimmer (nominal value adjustment)

4 External nominal value adjustment

5 Ramp connector (0.1 ... 5 sec.)
The rate of change of the input signal can be adjusted by the ramp connector.

6 Function connector
The overlap at the main spool of the proportional valve is jumped over during the switching process, in order to avoid idle time.

7 Diodes for direction control
Upper diode allows through + voltage
Lower diode allows through – voltage

8 Inverter stage for equalisation of the signal (+ voltage is changed to – voltage)

9 Voltage constant for solenoid a (fixed current generator)

10 Voltage constant for solenoid b (fixed current generator)
The voltage constant operates with impressed current, so that the current remains fixed when resistance changes (temperature change). P8 and P9 dead current; solenoid is pre-energised, to obtain short switching times.

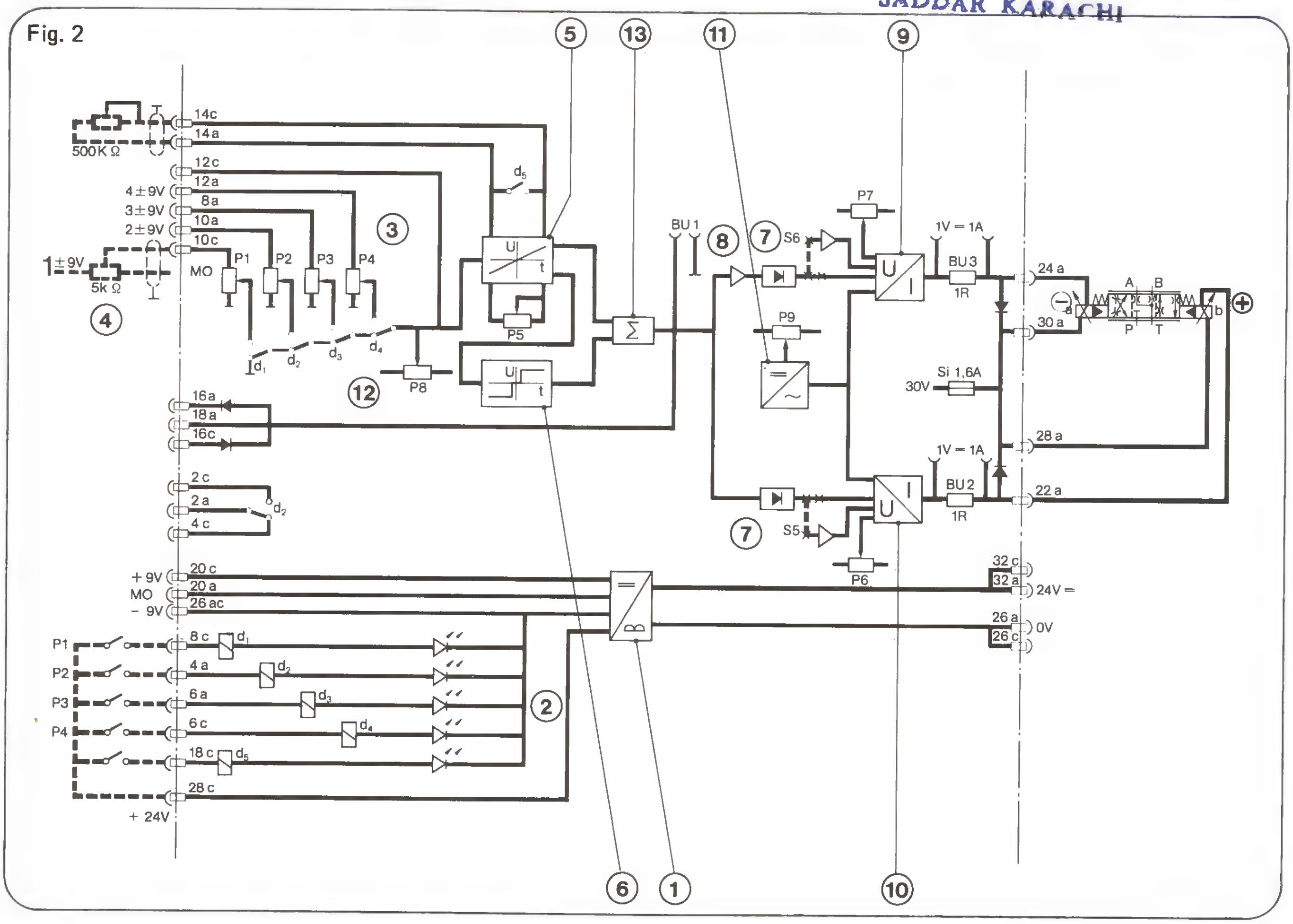
11 Dither generator, to improve the hysteresis P7 – dither current

12 P5 offset trimmer; setting of zero point, irregularities of the parts are balanced out. The setting is carried out at the factory.

13 Summator, for the addition of ramp and function generator

Acceleration and deceleration processes can be carried out very simply using these control electronics. The values required are put in via the electronics and are not dependent on hydraulic influences (oil viscosity).

Fig. 2



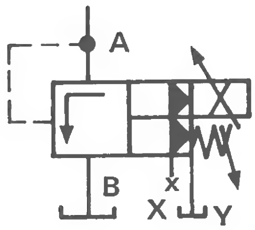
Proportional Valves

Pressure Relief Valve with Proportional Solenoid

Pilot operated pressure relief valve type DBE with maximum pressure protection

The valve comprises a pilot valve 1 with proportional solenoid 2, optionally with integral maximum pressure protection 3 and the main valve 4.

Symbol
type DBEM ... Y

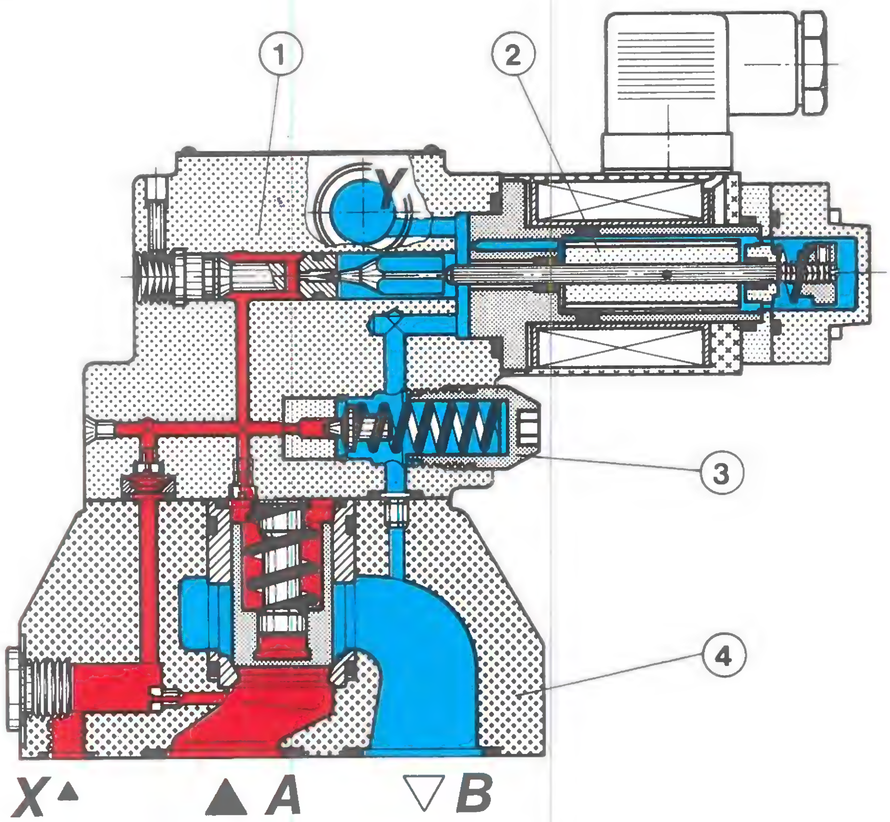


The basic function corresponds to that of the pilot operated pressure relief valve type DB described in the chapter "pressure relief valves". The difference is in the pilot valve. A proportional valve is used here in place of the spring and the pilot valve adapted accordingly.

The system pressure setting is related to the proportional solenoid current. Higher input current means greater solenoid force and thus a higher pressure setting.

Fig. 3 shows the maximum pressure protection valve, which prevents pressure exceeding the maximum permissible system pressure, if, for example, there is damage to the electronics and maximum electrical control current is developed.

Fig. 3



Proportional Valves

An electrical amplifier (type VT 2000) is used to control solenoids. Two trimmers to set the zero current and the maximum pilot current are in the amplifier.

The system pressure can be remote controlled by means of the potentiometer connected to the amplifier.

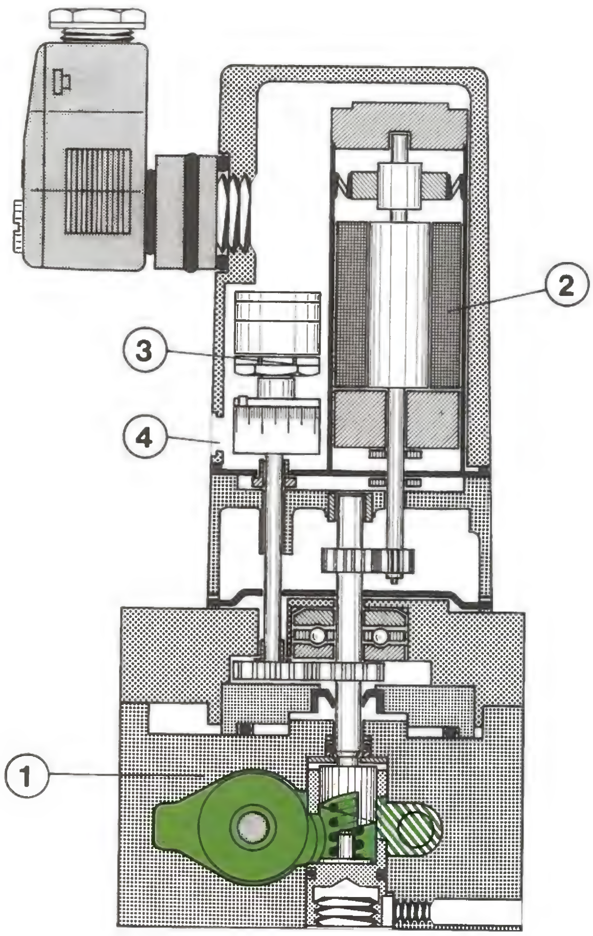
Important technical data:

Sizes	sizes 10, 25, 32
Operating pressure	up to 315 bar
Flow	up to 600 l/min

Along with this pressure relief valve, the pressure reducing valve (sizes 10, 25 and 32) is also available with proportional solenoid control.

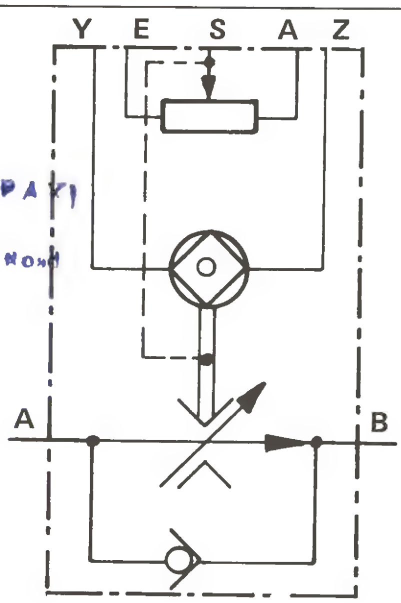
2 Directional Flow Regulating Valve with DC Servo Motor

Fig. 4



2 FRE

Symbol



This 2 way flow control valve, operated by means of a DC servo motor, should be mentioned in connection with the proportional valves previously described.

The hydraulic part, the 2 directional flow regulating valve 1, is identical to the flow regulating valve type 2FRM (see flow control valves). (fig. 4)

The flow section at the orifice port, i.e. operation of the metering spool, is adjusted by means of the DC servo motor 2.

Proportional Valves

The metering spool is connected to the servo motor by means of a gear unit.

A precision potentiometer 3, whose function is to feedback the orifice position, is engaged with the drive shaft of the metering spool.

The adjustment angle is 300° for the total flow range. It corresponds to a 10 part scale and can be read through a window 4 in the valve housing. Flow control valves type FRE are suitable for applications in regulating circuits, and also for remote and programmed controls. The adjustment drive is controlled by means of an operational power amplifier.

Servo Valves

The term "servo" is used a great deal. In general, it is used to describe the function where a small input (input signal) causes a large output (output signal).

The servo steering in a car must be best known, whereby the steering wheel is moved with only little force, but transmits great force to the wheels.

In servo hydraulics, the function is similar. A low power control signal, e.g. 0.08 watt, can give analgous control of several 100 kW. However, the servo valve does not only fulfil the requirements of analgous control. As opposed to proportional valves, it is generally used in electro-hydraulic regulating circuits, e.g. position regulating circuit (holding a position under load) or speed regulating circuit (maintenance of a given speed).

As the terms "control" and "regulate" are often used at random, we should like to define them briefly.

"Control" means in regulating technology that a nominal value is given and the actual value results from a fixed setting at the unit. The actual value is not monitored and therefore not corrected.

An example from hydraulic applications: the required control speed would result from a given flow through a flow control valve at a certain setting.

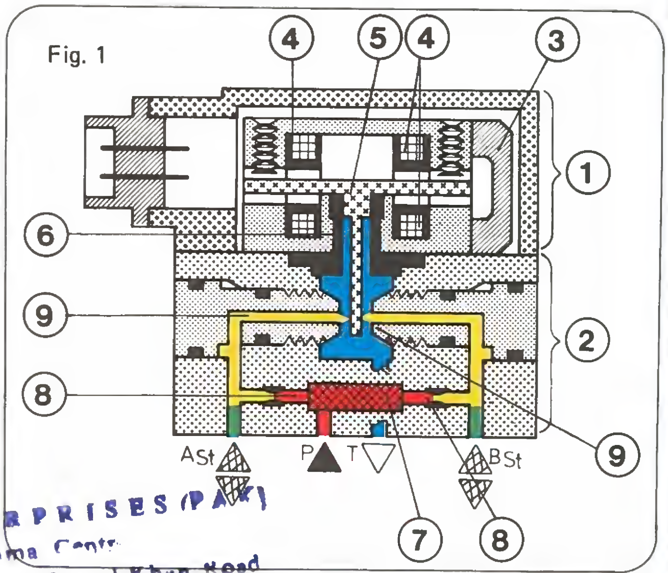
With "regulation" a nominal value (command variable) is given. The actual value resulting from this is measured continuously. If the actual value differs from the nominal value, a signal is formed from the difference, which influences the system so that the actual value is equated to the nominal value. The purpose of the regulation is therefore to equate the actual value to the command variable in spite of unfavourable influences.

In the servo valve, a small electrical input signal undergoes an analgous change into a hydraulic output signal (pressure, flow).

Directional Servo Valves

2 stage servo valves of different sizes
The first stage is shown separately, centre bottom

Fig. 1



The two stage directional servo valve comprises mainly the first stage (el. control motor 1 and hydr. amplifier 2) and the second stage.

Let us firstly take a closer look at the first stage. (fig. 1)

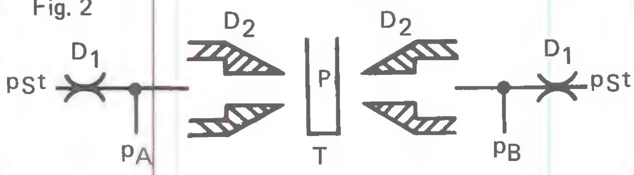
The control motor 1 (with permanent solenoid 3, control coils 4 and armature with flapper plate 5) changes a small current signal into a proportional flapper plate movement.
The armature and the flapper plate are one part, which is shock mounted to a thin-walled flexible pipe. At the same time, the pipe seals the control motor from the hydraulic fluid. The control motor is dry. The control coils are energised by means of a current signal and the armature deflected against the spring force of the pipe. The direction of deflection is determined by the polarity of the input current.

Servo Valves

The torque on the pipe and thus the deflection of the flapper plate is proportional to the control current. When the control current is shut off, the pipe (return spring) brings the armature and thus also the flapper plate back into centre position. The torque transmission of this type of control motor has the following advantages: no friction, low hysteresis and a sealing hydraulic medium/control motor (no magnetic field in the pressure medium).
The deflection of the flapper plate is transformed into a hydraulic value in the hydraulic amplifier 2. The jet flapper plate system is used as a hydraulic amplifier.

We shall now show the function, using a sketch of the basic principle (fig. 2).

Fig. 2



The system comprises 2 fixed jets D_1 and 2 control jets D_2. The control pressure p_{st} on both sides is reduced by means of jets D_1 and D_2 (= voltage decrease via el. resistance). If the jet sections are of equal size, an equal reduction in pressure via the jets occurs (e.g. p_{st} = 100 bar, p_A = 50 bar, T = 0).

When flapper plate P is deflected, the distances between the regulating jets change, for example, deflection to the left:

The distance of the plate to D_2 on the left decreases, to D_2 on the right increases. Pressure at p_A and p_B change in reverse accordingly. Pressure p_A increases, pressure p_B decreases. The pressure drop $p_A - p_B$ is taken as a useable signal.

The following diagram shows the change in pressure related to the inlet current (fig. 3).

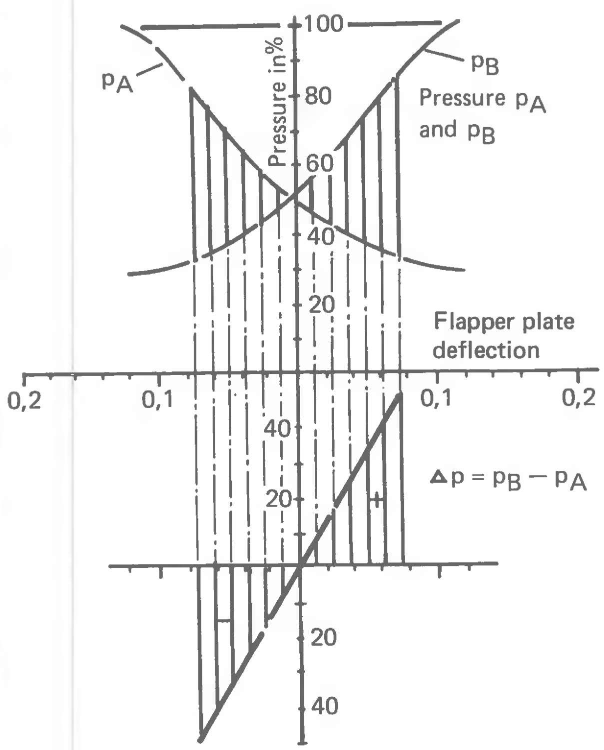
The jets are synchronised and adjusted to one another in such a way that the pressure difference shown results from a linear change in current.

The complete system is shown under item 2 in the sectional diagram (fig. 1).

Pilot oil is supplied from port P by means of a small protection filter 7 to the fixed jets 8 and further to the control jets 9.

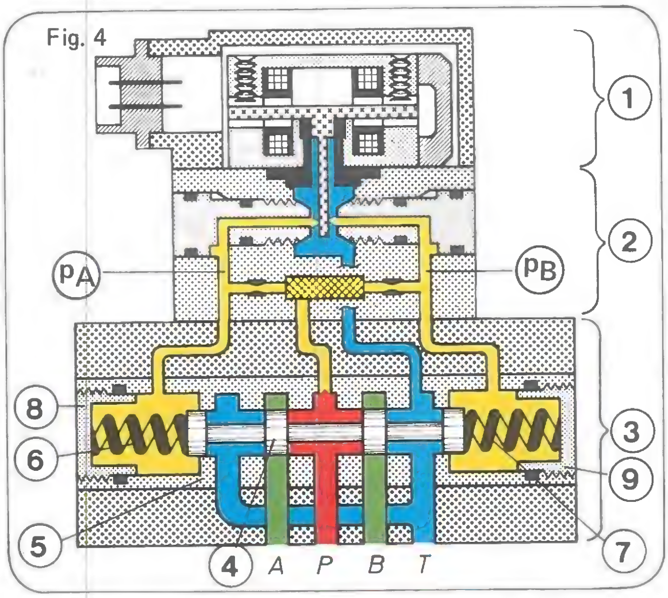
Pressure p_A and p_B is measured at ports A_{ST} and B_{ST} between the fixed and control jets.

Fig. 3



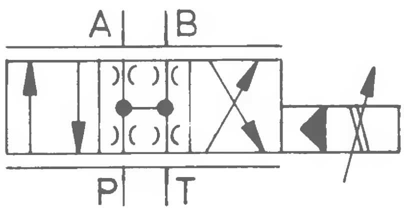
With this pressure difference, we now control the second stage spool (fig. 4).

Fig. 4



Servo valve, 2 stage, spring centered (barometric feedback)

Symbol



Servo Valves

Control motor 1	changes current i into distance s
Hydr. amplifier 2	changes distance s into pressure drop Δp
Second stage 3	changes pressure drop Δp to flow Q

The second stage is a control valve, whose control spool is in a wear-resistant bush 5 or directly in the housing if it is a simple model. The spool is centered between 2 pressure springs 6 and 7. The control and control bush are machined to suit one another in such a way, that almost zero overlap is achieved. Depending on the system, either a slight positive or a slight negative overlap is required in practice.

At inlet signal zero (i = 0) at the first stage, the flapper plate is in the centre. The pressure between the fixed and control jets is equal ($p_A = p_B$). The same pressure affects both sides of the control spool. The spool is in neutral position. With the overlap shown, ports P, A, B and T are blocked. If the flapper plate is now deflected to the left by means of a control signal, pressure rises in spring chamber 6, and pressure in spring chamber 7 decreases at the same time. The pressure drop pushes the control spool 4 to the right against spring 7, until the forces on both sides of the spool are in balance again.

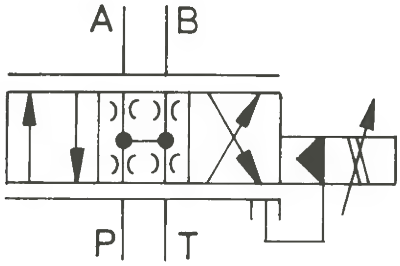
The spool has reached its position. As the pressure drop increases, i.e. greater input signal, the control spool moves further in one or other direction. The greater the spool stroke, the greater the opening section from P to A or B, the greater the flow, the greater the user speed.

The springs and thus the spool position related to the control lands can be adjusted externally by means of adjustment screws 8 and 9.

Mechanical Feedback (fig. 5)

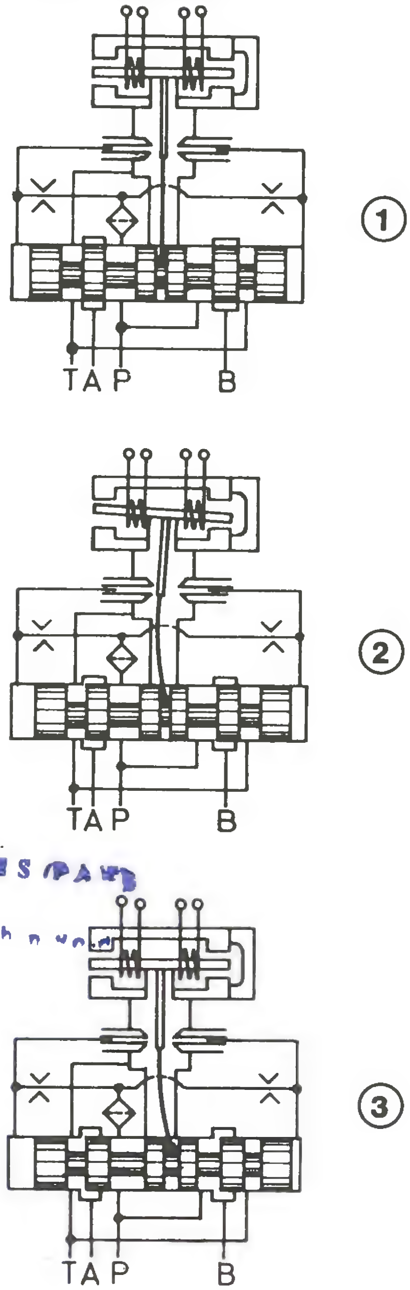
With mechanical feedback, the control spool is connected to the control motor of the first stage by means of a feedback spring. The spring exerts a torque against the control motor torque. When the required spool position is reached, the torque of the control motor is equal to the buckling torque of the feedback spring. The flapper plate is returned to the centre position. The torque equality leads to a pilot pressure balance and maintenance of the spool position. Spool stroke and flow are therefore proportional to the input current.

Symbol



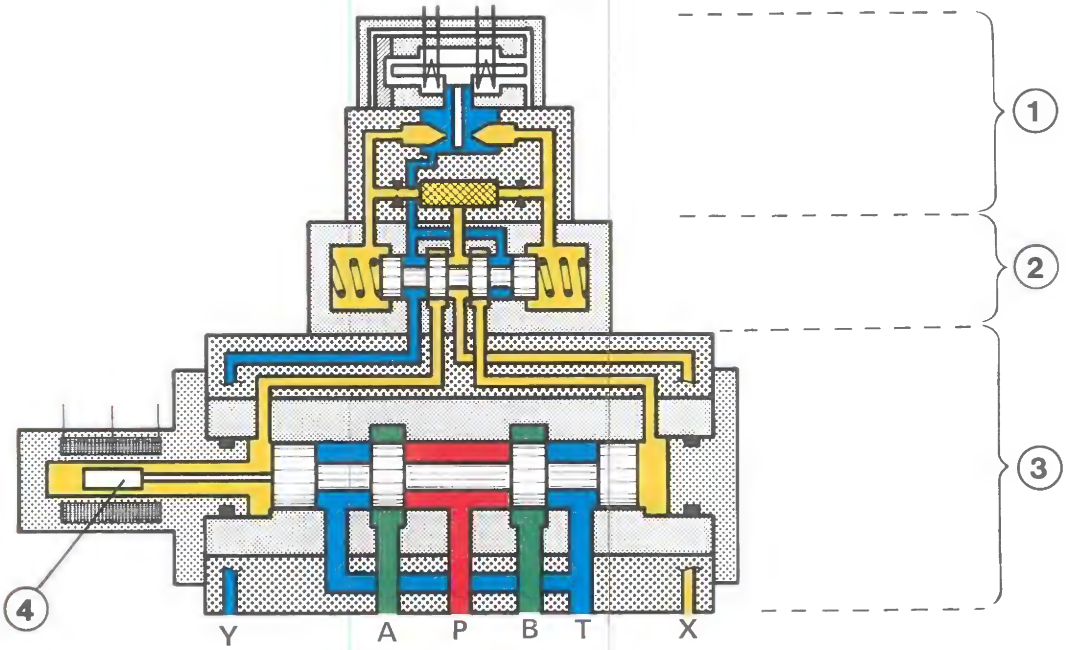
1 Valve in neutral position
2 Flapper plate is deflected
3 Spool has reached its position

Fig. 5



Servo Valves

Fig. 6



El. feedback (fig. 6)

1 first stage, 2 second stage, 3 third stage, *3 stage directional control valve with el. feedback*

The deflection of the pilot spool is monitored by a position measuring system and compared with the nominal value by an amplifier. Inductive directional pick-ups, which create an electrical output signal proportional to the position of the spool, are widely used for measuring the spool position.

The core of the pressure-tight directional pick-up is flexibly fixed to the control spool. When the spool is deflected, a voltage drop is created by means of the core in the AC pick-up coils. This signal is proportional to the stroke and is evaluated by suitable electronic units, before being fed to the servo valve as the required deviation.

Fig. 6 shows a 3 stage directional servo valve. The second stage serves to control the main spool of the third stage. As with directional spool valves, this is necessary for larger oil flows, to achieve corresponding switching or adjustment times.

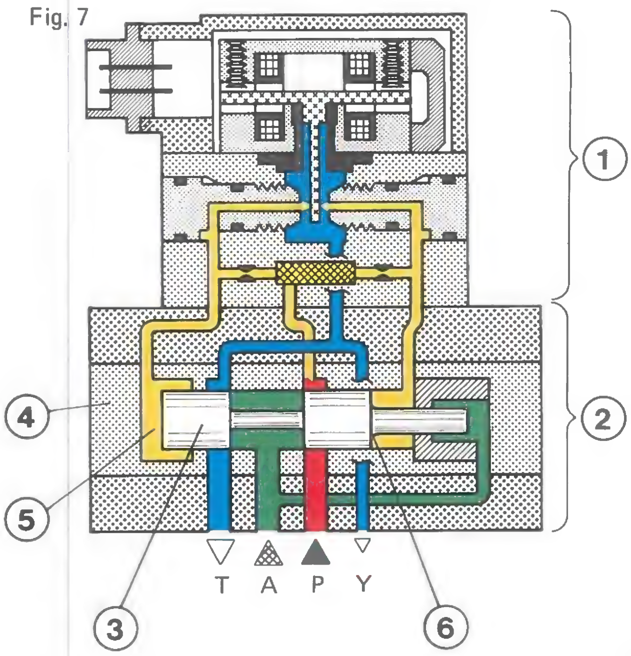
Important technical data:

up to 315 bar	sizes 8 – 32
up to 700 l/min	connection dimensions to DIN 24 340

Pressure Servo Valve

The 3 directional pressure servo valve shown in fig. 7 serves to reduce pump pressure to a user pressure proportional to an electrical input signal.

Fig. 7

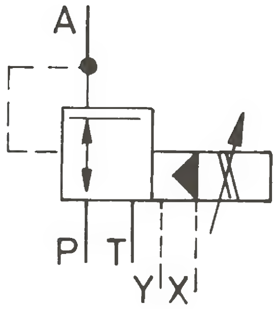


1 first stage

2 second stage with pressure regulating spool

Servo Valves

Symbol



The first stage, which we already know from the directional servo valves, is used as the control element. The regulating spool 3 is fitted in a control sleeve 4. At input signal zero, half the control pressure affects the left-hand full area face 5 and half area face 6 (annulus area) lying opposite. The remaining half area face of the spool is connected to port A, which is to be regulated, and serves as a hydraulic feedback.

The control spool is pushed to the right, until the pressure in the user line A has reached half the system pressure. When the baffle plate is deflected, the user pressure decreases or increases proportionally, until the spool is once again in equilibrium. The pressure to be regulated at A is in the range
$p_T \leqslant p_A \leqslant p_{ST}$

All valve sizes can be externally adjusted. This means that the relationship between input current and control pressure can be set so that at input signal i = 0, the pressure at port A = 0.

Notes

Programme Summary

- Accessories
  - Filters
    - Suction filters
    - Pressure filters
      - p_{max} = 420 bar; Q_{max} = 660 l/min; filtration = 1, 5, 10, 20 μm
    - Return line filters Single and double filters
      - p_{max} = 30 bar; Q_{max} = 6500 l/min; filtration = 10, 20, 25 μm
    - Filler / breathers
      - Air filter = 10, 20, 40 μm; Rate of air flow = max. 2.6 m³/min; Oil filter = mesh width 0.5 mm
  - Pressure switches (hydro-electric)
    - Piston type pressure switches
      - Threaded connections, p_{max} = 500 bar; Subplate mounting p_{max} = 350 bar; Switching pressure differential pressure related
    - Bourden tube type pressure switches
      - Threaded connections, p_{max} = 400 bar; Constant switching pressure differential; Adjustable switching pressure differential
  - Pressure indicating units (multi-circuit gauge isolator)
    - with built-in pressure gauge
      - p_{max} = 400 bar; 6 measuring ports
    - without pressure gauge
      - p_{max} = 315 bar; 6 or 5 measuring ports
  - Pressure gauge isolator valves
    - p_{max} = 300 bar; threaded connections, subplate mounting
  - Pressure control units
    - with 1 or 2 pressure relief valves
      - p_{max} = 400 bar; threaded connections
  - Heat exchangers
    - Oil/air cooler, model for coupling housing
      - p_{max} = 8 bar; Q_{max} = 15 l/min
  - El. accessories
    - Immersion heaters
    - Thermostats
    - Thermometers
    - Float switches

Programme Summary

- Hydraulic accumulators
  - Bladder accumulators
  - Membrane accumulators
  - Piston accumulators
- Methods of connection for hydraulic units
- Pipes, hoses, couplings, flanges, etc.
- Subplates
- Manifold blocks
- Control blocks
  - only line connection
  - Line connection and wear-resistant housing
  - Line connection and "true" housing
- Control blocks with integrated circuit (compact hydraulics)
  - Tower and block stacking assemblies
    Tracing systems, feed controls, grinding machine controls
    Lowering modules, fuse protection modules, deep drawing modules, small press modules
    Pressure controls, direction controls by means of logic elements, 3 directional flow controls
    Upper and lower cylinder control
- Valve stacking assemblies
  - Vertical stacking assemblies
  - Horizontal stacking assemblies

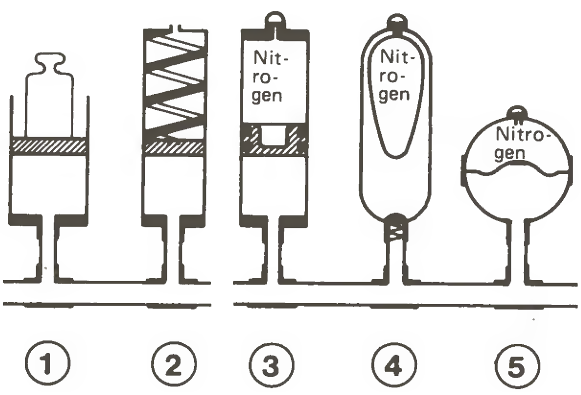
Hydraulic Accumulators

A hydraulic accumulator serves to take into store a volume of fluid under pressure and to release it again, as required. The pressure accumulator can carry out many tasks in a hydraulic circuit.

- As **fluid reserve**, where a large quantity of fluid may be required at short notice in a hydraulic system. The hydraulic pump is not designed for the maximum flow required for a short time only. It has a lower flow volume and fills the accumulator, if, during the working cycle, the volume of fluid required for the system is lower than the pump flow. If the maximum volume is then required, the difference between this and the pump flow volume is taken from the accumulator. The accumulator therefore helps to avoid the use of a large pump with high drive power, suitable for the short-term high power requirement.
- As an **emergency unit**, to stop an operation which has already started, should there be any damage to the pump or its drive.
- As **leakage compensation**, to make up leakage losses and thus maintain pressure over a long period.
- To **balance out the volume** at temperature changes, e.g. for a closed system.
- To **break down pressure peaks** during switching processes.
- To **cushion vibrations**; decrease of pressure amplitudes on pumps.
- For the **recovery of brake energy**.

Various models of accumulator are available:

Fig. 1



1 Weight accumulator
2 Spring accumulator
3 Piston type accumulator
4 Bladder accumulator
5 Membrane accumulator

Weight and spring type accumulators are of practically no importance for industrial applications.

The gas pressure accumulator is used most frequently. The actual accumulation of pressure energy is undertaken by the compressible gas (nitrogen). One differentiates between piston, bladder and membrane accumulators.

Piston Accumulators

They are suitable mainly for large volumes and large discharge quantities. The gas and fluid are separated by a free-moving piston ("flying piston"). The piston runs in a cylinder tube and provides a seal between the gas and the fluid by means of rings. The maximum pressure ratio, i.e. the ratio gas pressure to maximum operating pressure, is 1 : 10.

Membrane Accumulators

They are used for small volumes, for example, to absorb shocks, cushion vibrations and for pilot circuits. The membrane, which is generally semi-spherical, divides the two medien and arches to the fluid side.
The maximum pressure ratio is again 1 : 10.

Bladder Accumulators

Hydraulic Accumulators

Fig. 2

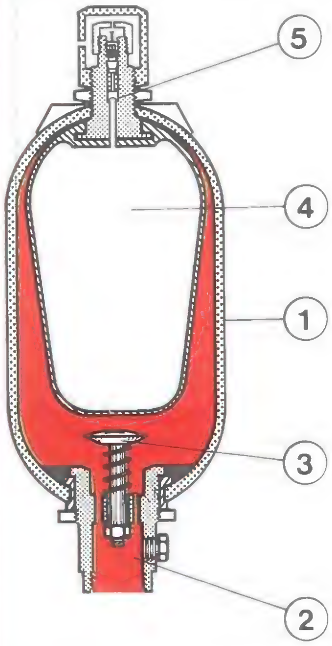
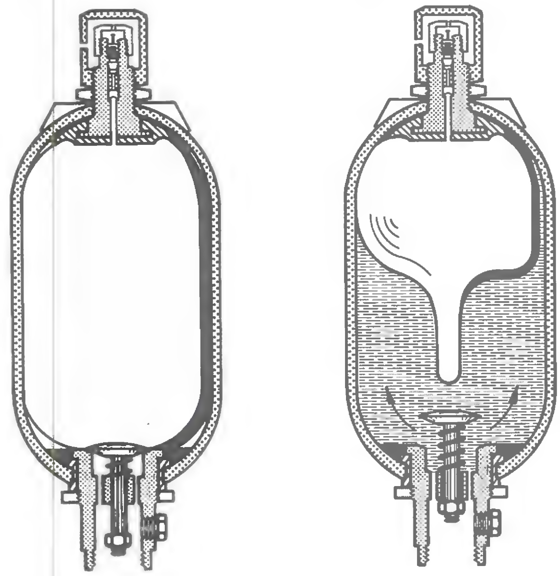


Fig. 3

Fig. 4



Symbol

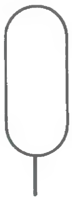
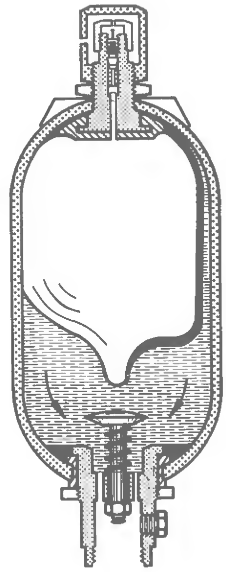


Fig. 5



It is distinguished by its absolute sealing feature, very short response time and very low inertia operation.

In a bladder accumulator, the nitrogen and fluid are separated by a closed flexible bladder. The gas is inside the bladder.
The maximum pressure ratio is 1 : 4.

The bladder accumulator (fig. 2) comprises a steel container 1 with fluid connection 2, plate valve 3, accumulator bladder 4 and gas valve 5.

The accumulator bladder 4, initially stressed with gas via gas valve 5, completely fills the steel container and closes the plate valve (fig. 3). The plate valve prevents the bladder from coming out of the container and also protects it from damage.

If the pressure in the hydraulic system becomes equal to the initial gas stress, fluid then flows into the accumulator by means of the plate valve and compresses the nitrogen in the bladder (fig. 4). The gas volume is decreased by the fluid intake volume. As the fluid drains, the accumulator bladder increases in size again (fig. 5). The gas pressure and also the pressure in the system follow the gas laws:

$$p \cdot V^n = \text{constant.}$$

p = gas pressure

V = gas volume

If the conditions change very slowly, causing exchange of heat, one talks of the **isothermal change of conditions.**

The gas temperature remains constant.

The power n = 1.

Hydraulic Accumulators

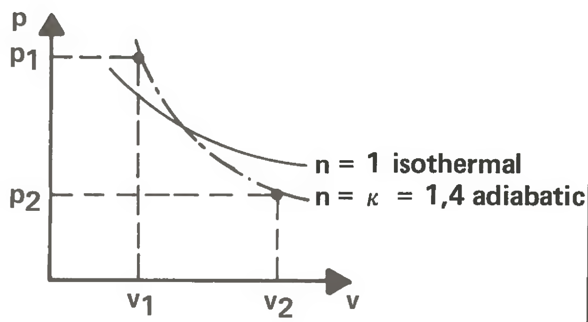
With an **adiabatic change in condition,** there is no change in heat. This means that an exchange of heat between the gas volume in the accumulator bladder and its immediate surroundings is not possible, when considering a pressure accumulator system. This conditions occurs, if the compression or expansion processes occur very quickly.

The power $n = \kappa - 1.4$

gas equation $p_1 \cdot V^n = p_2 \cdot V^n$

Representation of the process in P-V diagram (fig. 6)

Fig. 6

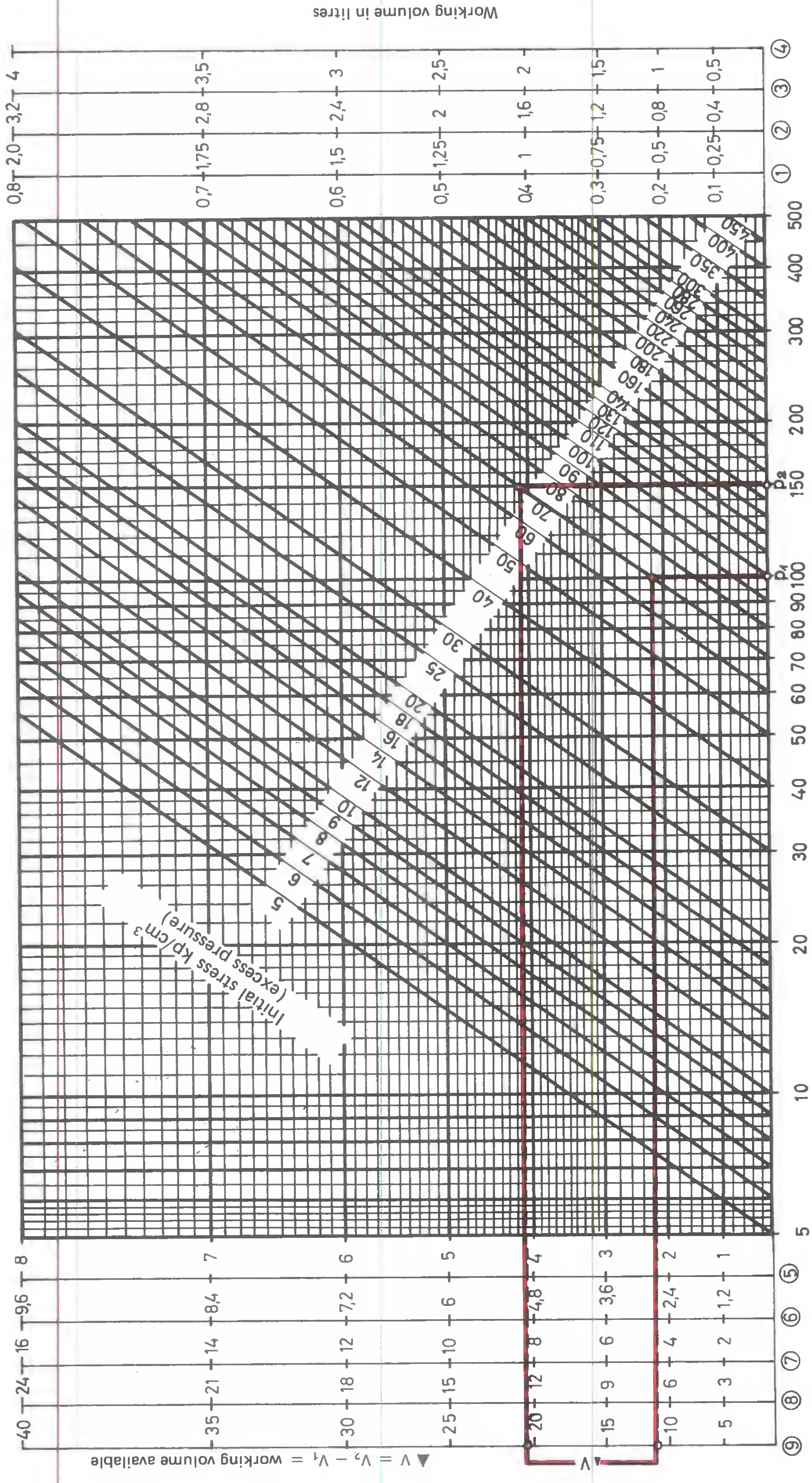


In practice, the change in conditions will lie between adiabatic and isothermal, depending on the drain speed. This is called a **politropic change in conditions**. The power n lies between 1 and 1.4. ($1 < n < 1.4$).

The following quantities for the accumulator are calculated by means of the gas equation: nominal volume, working volume available and prefill pressure related to the minimum and maximum operating pressure. These data can also be taken direct from the diagram, using the power performance curves (fig. 7).

Hydraulic Accumulators

Fig. 7



P_1 = min. operating pressure, P_2 = max. operating pressure in the hydraulic unit kp/cm³ (excess pressure)

The diagram is based on the gas laws: $V_0 = V_1 \left(\frac{p_1}{p_0}\right)^{\frac{1}{\kappa}}$ or $p_1 = p_0 \left(\frac{V_0}{V_1}\right)^{\kappa}$

(κ – nitrogen = 1.4

p_1 and p_2 in kp/cm³ absolute pressure)

For de-stressing under 2 minutes, when temperature between gas and pressure fluid is not balanced out.

Diagram: Messrs. Hydac Gesellschaft für Hydraulic-Zubehör mbH

Nominal size of the accumulator in litres

① = 1 / ④ = 5 / ⑦ = 20
② = 2,5 / ⑤ = 10 / ⑧ = 32
③ = 4 / ⑥ = 12 / ⑨ = 50

Adiabatic performance curve

Hydraulic Accumulators

The prefill pressure initial gas stress of the accumulator should lie between 0.7 – 0.9 of the minimum working pressure.

$$p_0 \leqslant 0.9 \times p_1$$

p_0 = initial gas stress

p_1 = minimum operating pressure

p_2 = maximum operating pressure

This should prevent the accumulator bladder from operating constantly in the oil valve range, which could cause damage.

The smaller the pressure difference between p_2 and p_1, the larger is the accumulator, related to a pre-stated working volume.

If accumulators are used in hydraulic systems, certain instructions must be heeded.

All pressure accumulators are subject to the safety instructions of the professional associations (UVV 13.5 pressure containers).

A few important points from these instructions:

- Each pressure container must have a suitable pressure gauge, indicating the operating pressure. The highest operating pressure permissible must be clearly marked.
  (This refers to an additional pressure gauge.)
- A suitable safety valve must be available for each pressure container. The setting must be secured against alteration by unauthorised people (control seal).
- The safety valve must not be lockable.
- Easily accessible isolating devices must be available in the pressure supply lines, as near as possible to the pressure container. It must be possible to isolate each container (accumulator).

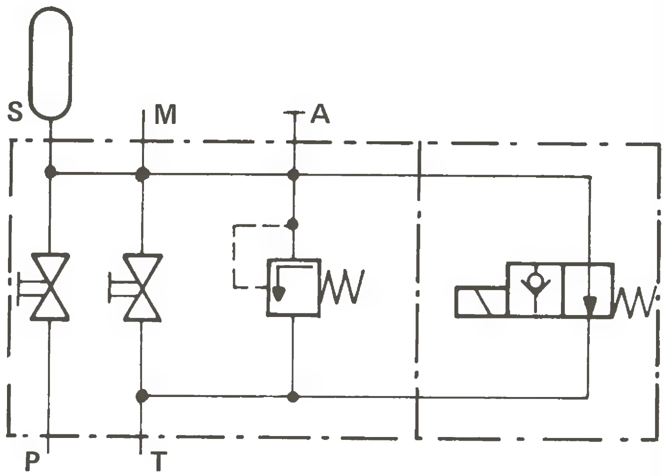
- Test requirement for pressure containers:

Group A: maximum operating pressure over 0.5 bar pressure litre product $p \cdot 1 \leqslant 200$
Accumulator need not undergo test.
p = maximum operating pressure of the accumulator (not the complete unit)
l = volume of the pressure chamber (accumulator size)

Group C: maximum operating pressure over 0.5 bar pressure litre product
$p \cdot 1 > 200$ to 1000
Accumulators must be tested before first commissioning.

Group D: maximum operating pressure over 0.5 bar pressure litre product $p \cdot 1 > 1000$
Accumulators must be tested before first commissioning and at regular intervals thereafter.

The safety and shut-off block in the following diagram corresponds to the first points mentioned.



S = accumulator connection
M = pressure gauge connection
P = pump connection
T = tank connection
A = test connection

Electrical unloading is also possible, as shown in the circuit on the right.

Notes

Accessories

The term "accessories" actually leads to a false impression, when considering the importance of the equipment in this section.

Just as in the case of energy transformers or control elements, they are important elements for certain control and monitoring functions, or elements necessary for a safe functioning hydraulic system.

Filters

above:
left: pressure filter
right: return line filter

right: return line filter (as double filter) with el. visual clogging indicator

The reliability of hydraulic units depends on the cleanliness of the system, i.e. on the filtration. The filter serves to reduce the level of dirt in a pressure medium to a reliable level, and thus to protect the individual elements from too much wear.

Various factors play a part:

– type of dirt particle (size, condition)

– number of dirt particles

– speed at which the pressure fluid flows into the individual elements

– system pressure, pressure drop

– tolerances, constructional conditions.

Test on fluids have indicated a connection between the amount of contamination, the size and the number of particles.

The degree of contamination is divided into 7 classes according to SAE standards:

Size of particle μm	No. of particles in 100 cm^3/classes						
	0	1	2	3	4	5	6
5 – 10	2700	4600	9700	24000	32000	87000	128000
10 – 25	670	1340	2680	5360	10700	21400	42000
25 – 50	93	210	380	780	1510	3130	6500
50 – 100	16	28	56	110	225	430	1000
100 –	1	3	5	11	21	41	92

Accessories

The particles, tiny pieces of dirt, are measured in microns [μm], the millionth part of a meter. Filtration is also stated in microns.

Absolute filtration: this refers to the value, corresponding to the diameter of the largest particle which can pass through the filter.

Filter Element, Filter Material

Filter element made of different materials left to right: wire mesh, paper, metal fibre

There is a star-shaped fold in the filter material. It is thus possible to achieve a very large filter area with a small size element and good stability.

Wire Meshing

A stainless steel mesh is used here.

Paper

The element is made of paper fibre. The filtration is 10 μm. Along with the pressure stable supporting tube and the star-shaped folding, the paper element guarantees relatively good inherent stability. Paper filters cannot be cleaned. They must be thrown away and are therefore used for flushing processes or commissioning of a unit.

Metal Fibre

Fibres are used as filter material. The metal fibre element has several advantages:

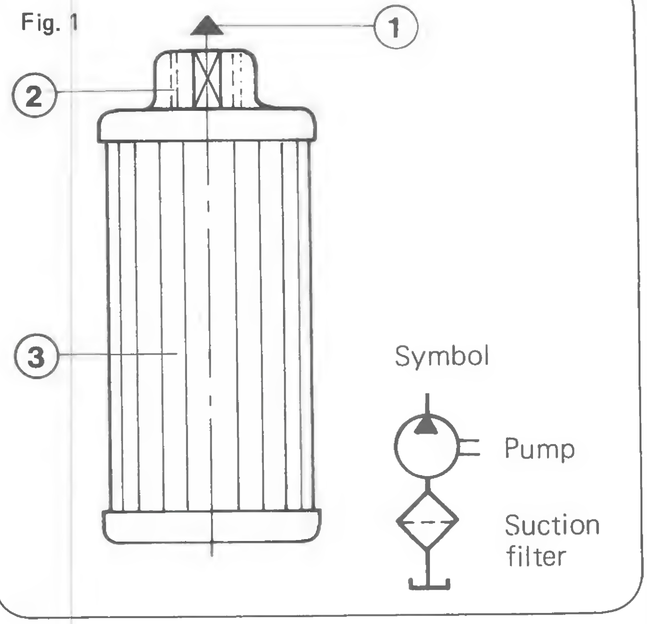
- multiple dirt pick-up capacity on the same filter area
- long service life due to deep filtering
- not related to temperature
- high permissible pressure drop
- good inherent stability

Various system filters are available, which differ from one another as follows, depending on their arrangement in a hydraulic circuit:

Suction Filters

The suction filter (fig. 1) is fitted in the suction line of the pump.

Fig. 1

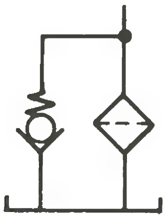


The filter element is fitted with a threaded port 2. The fluid is sucked out of the tank through the filter element 3, so that only filtered oil reaches the system 1. However, a disadvantage is that it is not easily accessible and maintenance is therefore difficult. A suction filter also makes the pump suction more difficult. Special attention must be paid to this point, as some pumps should not be fitted with this type of filter. The filtration is generally > 100 μm.

The suction filter elements can also be fitted with a by-pass valve, so that no suction difficulties can occur with a dirty element or at a cold start. The cracking pressure is 0.2 bar.

Accessories

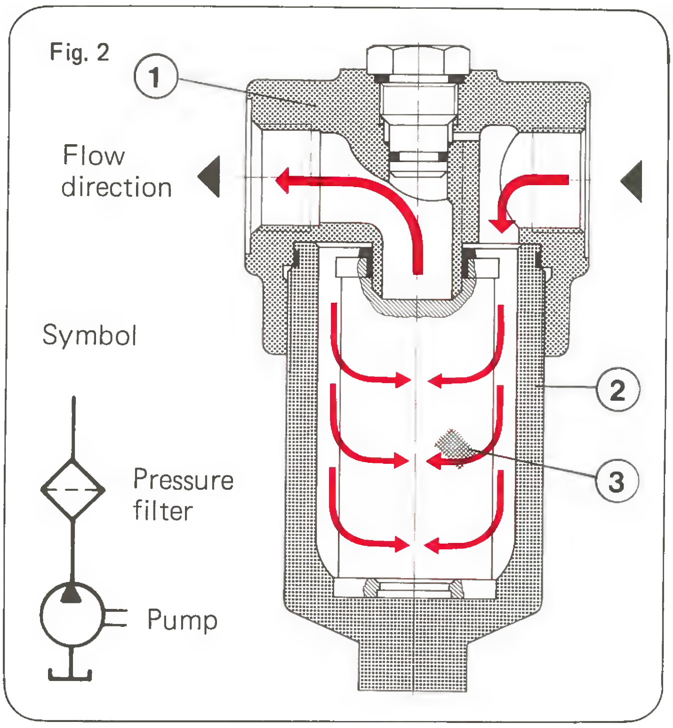
Suction Filter with By-pass Valve



Pressure Filter

The pressure filter is fitted in the pressure line of a hydraulic circuit. This can be, for example, at the pressure port of the pump, in front of a servo valve or in front of a flow control valve, set at a very low flow. However, The filter is usually fitted directly in front of the unit to be protected.

Fig. 2



The pressure filter (type DF) shown in fig. 2 is suitable for fitting in pressure lines. The filter comprises a filter housing with filter head 1, screw-in filter drum (dirt catching drum) and the filter element 3. As the pressure filter is exposed to the maximum operating pressure, it must be correspondingly stable. This filter is designed, for example, for a permissible pressure drop of 315 bar.

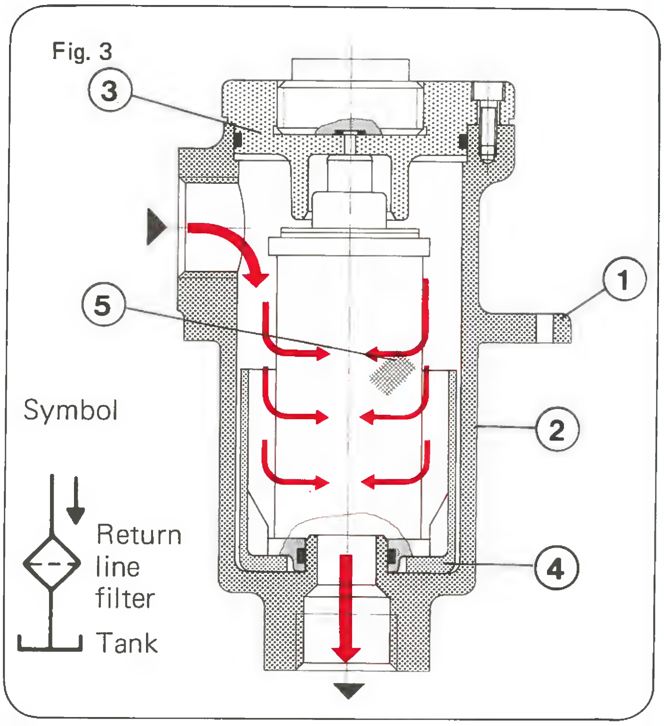
Important technical data:

Operating pressure:	up to 420 bar
Flow:	up to 330 l/min at $\Delta p = 0.8$ bar
Filtration:	1, 5, 10 μm

Return Line Filters

The return line filter is the type most used. The filter is fitted in the return line. This means that fluid coming from the system is filtered and flows back to tank.

Fig. 3



The filter is available for fitting in the oil tank (fig. 3) or for direct line mounting.

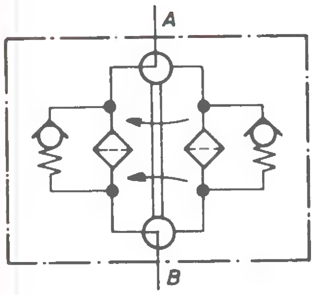
The filter shown in fig. 3 is fixed on to the tank cover with the mounting flange 1. The housing 2 with the filter outlet goes directly into the tank. One advantage of this type of filter is easy accessibility and therefore ease of service.

It is also important that the filter element 5 is surrounded by a dirt catching tray 4. The tray is removed with the element. The dirt cannot therefore flow away into the oil tank. Double filters are also used, in order to avoid standstill, caused by element changing or filter maintenance.

Two filters are fitted parallel to each other. Standstill of the unit can be avoided by switching to the second element, while the first element is removed.

Accessories

Symbol
Double Filter



Important technical data:

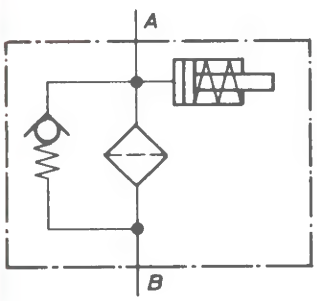
Operating pressure:	up to 30 bar
Flow:	up to 1300 l/min (filter for tank mounting)
	up to 3900 l/min (filter for line mounting)
Filtration:	10 and 20 μm

Clogging Indicators

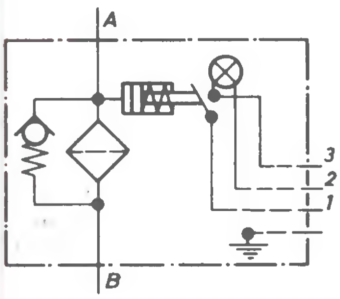
The degree of filter contamination can be determined indirectly by means of the flow resistance.

Pressure in front of the filter element affects a spring loaded spool. As pressure rises, meaning also as contamination increases, the spool is pushed against the spring. This piston path can then either be visible directly, or can be transferred to an electrical and visual indicator, by means of electrical contacts.

Symbol



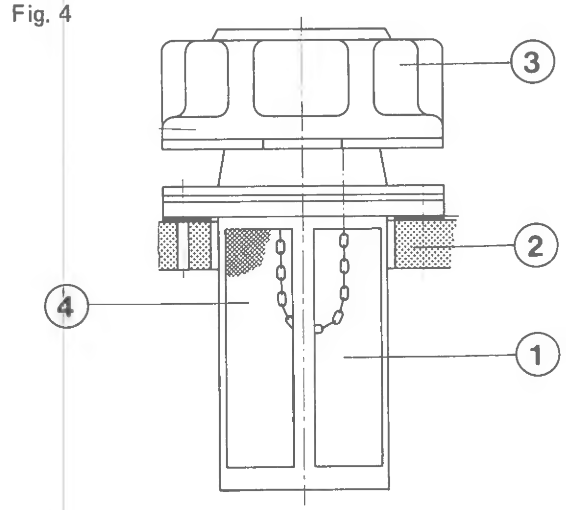
Filter with visual (mechanical) clogging indicator



Filter with el. visual clogging indicator

Filter/Breather

Fig. 4



Symbol

The filler/breather type ELF is for mounting in the tank (fig. 4).

It serves two purposes:

– as a filling filter: When the tank is being filled with fluid, the filter prevents large particles of dirt entering the tank and the system. Filling up should therefore be carried out basically using a filling filter.

– as a bleed filter: Where the fluid level varies, for example, due to differential users, the amount of air must change. The air flowing into the tank is filtered.

The filter 1 is fitted on to the tank cover 2. To fill the tank, the cover 3 with bayonet catch is removed. The cover is secured with a chain.

Accessories

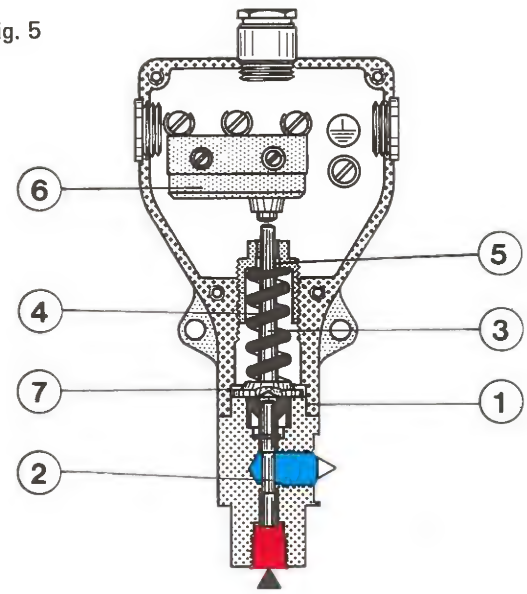
Pressure Switches

Hydr. - el. pressure switches
right: piston pressure switch HED 1
left and below: bourdon tube pressure switch HED 2 and 3

Hydro-electric pressure switches serve to switch an electrical circuit on and off, related to pressure. The pressure switch can be used as a control unit or also for monitoring purposes, i.e. by means of visual (lamp) or accoustic (bell) indicator. We shall look at two types of pressure switch in detail:

Piston Pressure Switch

Fig. 5



Piston pressure switch type HED 1

Symbol

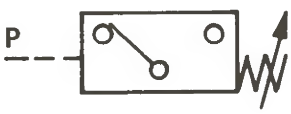
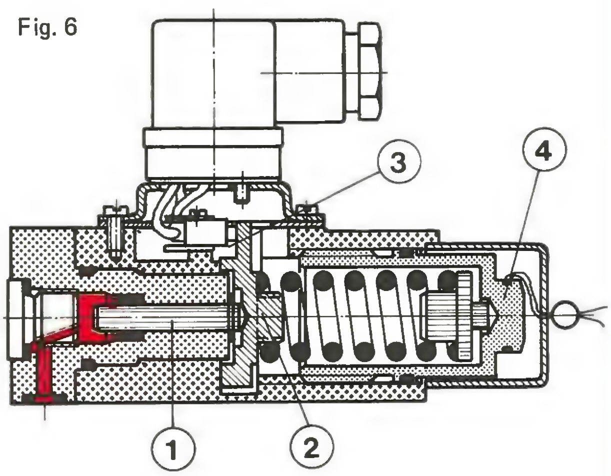


Fig. 5 shows pressure switch type HED 1. The piston 2, plunger 3 with spring 4, adjustment screw 5 and micro switch 6 are fitted in a housing 1.

The pressure to be monitored affects the piston 2, which pushes against the spring 4 by means of the plunger 3. The spring force can be adjusted by means of the adjustment screw 5. If the pressure force at the spring exceeds the spring force, the piston is pushed against the spring. The plunger transmits the movement to the micro switch. A mechanical stop 7 protects the micro switch from damage at excess pressure.

Fig. 6



Piston pressure switch type HED 4

Another version of the piston switch is shown in fig. 6. The piston 1 affects a spring retainer 2. The spring retainer is fitted with a lug to operate the micro switch. The initial tensioning of the spring and thus the switching point is carried out via the sleeve 4. An advantage of this construction is that the hydraulic porting is also possible with subplate mounting

On piston pressure switches, the switching pressure difference is pressure related. They are suitable fo opening and closing. Two switches are required for 2 switching points.

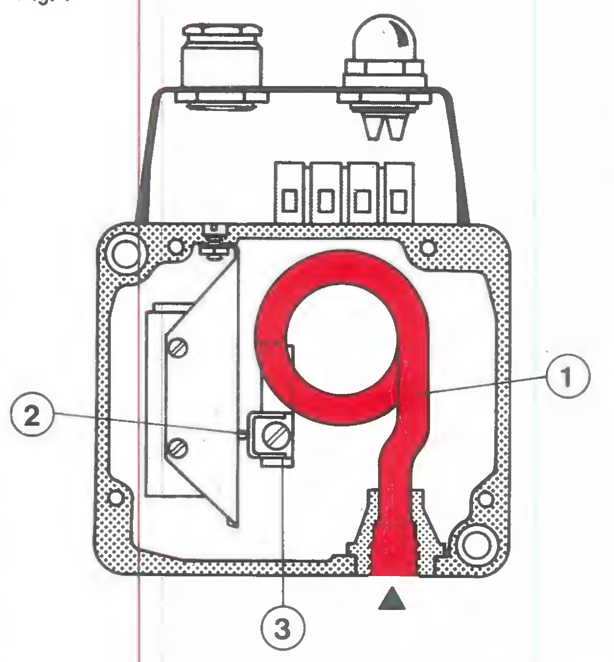
Important technical data:

max. set pressure:	HED 1	= 500 bar
	HED 4	= 350 bar

Accessories

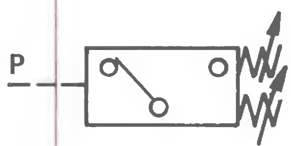
Bourdon Tube Pressure Switch

Fig. 7



Bourdon tube pressure switch type HED 2

Symbol (pressure switch with 2 micro switches)



On this type of pressure switch (fig. 7) the pressure affects a bourdon tube 1. According to the amount of pressure, the bourdon tube bends upwards and operates the micro switch 2. The switching point is set over the difference of the micro switch to the switching lever 3. Here again, a mechanical stop protects the micro switch from damage at excess pressure.

The pressure switch type HED 2 has a lockable adjustment knob, which is used for external adjustment of the lower switching pressure. The upper switching pressure results from the constant pressure drop over the total adjustment range.

The pressure switch type HED 3 (see photo) is fitted with 2 micro switches. The upper and lower switching pressures are set externally using knurled screws.

Bourdon tube pressure switches can be used not only for mineral oils, but also for gas or air.

Important technical data:

max. pressure setting: 400 bar

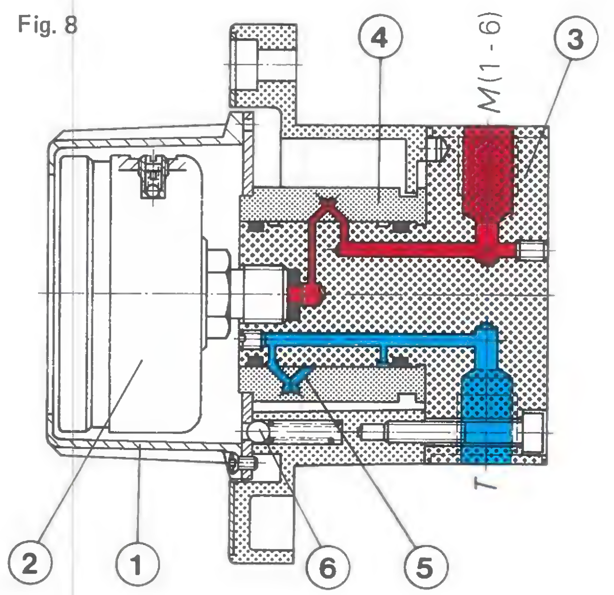
Multi-Circuit Gauge Isolator

Multi-circuit gauge isolator type MS 2

Multi-circuit gauge isolators type MS are rotary slide valves. They make it possible to select 5 or 6 measuring points in a hydraulic unit to test the operating pressure there.

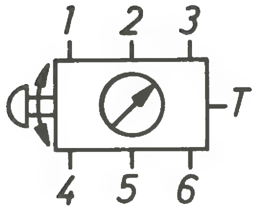
Multi-Curcuit Gauge Isolator Type MS 2

Fig. 8



Accessories

Symbol



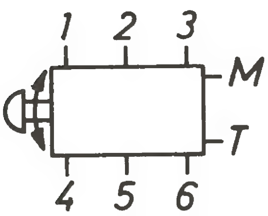
A glycerine cushioned pressure gauge 2 is built into the adjustment knob 1 of this isolator valve. The 6 measuring ports M are arranged at the edge of the housing. When the adjustment knob is turned, and the sleeve 4 coupled with it, one measuring port is always connected with the pressure gauge, as shown in fig. 8.

There are zero positions between the measuring ports for unloading the pressure gauge. The pressure gauge is thus connected to the tank (port T) by means of the bore 5 in the sleeve. The detent 6 fixes the set measuring or zero position.

An arrow on the edge of the adjustment knob shows which measuring port is connected to the pressure gauge.

Another type of multi-circuit gauge isolator, type MS4 for 6 measuring points without built-in pressure gauge is available. The measuring points are connected to the externally arranged pressure gauge by turning and then pushing in the rotary knob. When the rotary knob is released, it returns to neutral position and the gauge is thus unloaded.

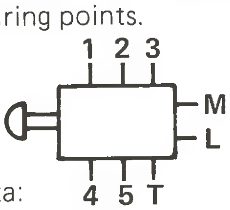
Symbol for MS4



M = pressure gauge

Multi-circuit gauge isolators type MS5 are operated in the same way as the MS2, but have no pressure gauge and have 5 measuring points.

Symbol for MS5



Important technical data:

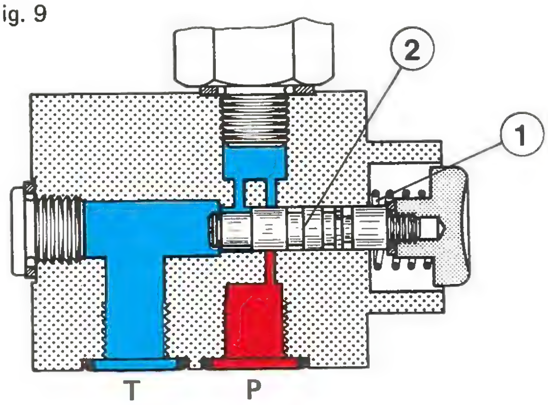
operating pressure:	315 bar		
measuring points:	6	5	5
	MS2	MS4	MS5

Pressure Gauge Isolator Valve and Pressure Control Unit

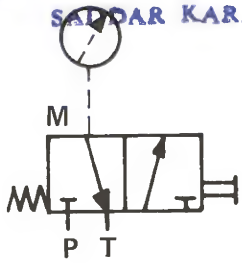
left: pressure control unit
right: pressure gauge isolator valve

Pressure Gauge Isolator Valve

Fig. 9



Symbol



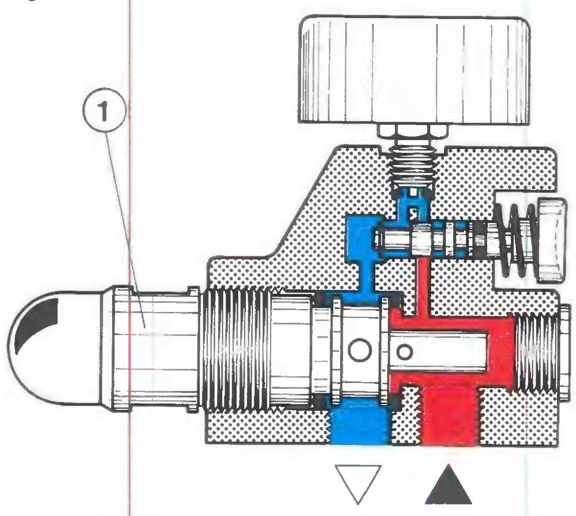
The pressure gauge isolator valve is a 3 way slide valve with push button operation. It serves to check the operating pressure periodically.

The valve has 2 switching positions. In zero position which is achieved by spring return 1, the spool 2 blocks the pressure port and the pressure gauge is connected with the tank.

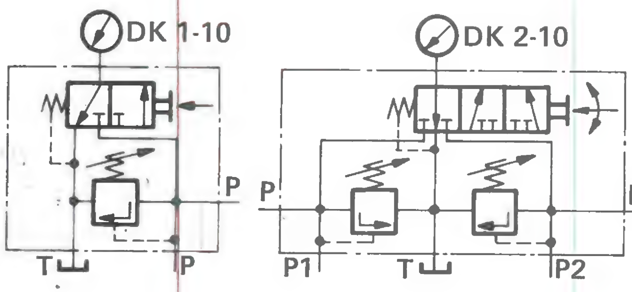
In switching position, there is free flow from the pressure port to the pressure gauge and the connection to tank is blocked. The pressure gauge can be screwed to the valve housing or arranged separately.

Pressure Control Unit

Fig. 10



Symbol



Pressure control units type DK are combination units. They serve to limit pressure in a system and to indicate pressure in one (DK1) or two (DK2) systems.

They comprise:

DK 1: housing, 1 pressure relief valve, pressure gauge and 3/2 directional control valve (pressure gauge protecting valve).

DK 2: housing, 2 pressure relief valves, pressure gauge and 4/3 directional control valve (pressure gauge protecting valve).

The pressure relief valve 1 is a direct operated pressure control valve type DBD (see chapter on pressure control valves).

In neutral position, the 3/2 directional valve (on DK 1) or the 4/3 directional valve (on DK 2) separates the pressure gauge from the system. This prevents the pressure gauge being constantly under pressure.

With the DK 1, the system is connected to the pressure gauge by pushing and with the DK 2 by turning to the right or left and then pushing.

Important technical data:

size pressure control valve:	size 10
operating pressure:	up to 400 bar

Heat Exchanger (Cooler)

In a hydraulic system, part of the output is transformed into heat at various points (lines, valves), i.e. the fluid heats up. If the heat radiated from the tank is too low, the induced temperature lies above the desired operating temperature, due to the amount of heat supplied and radiated. The fluid must be cooled. The cooler ensures that a certain fluid temperature is not exceeded.

There are two different types:

– Air Cooled Heat Exchanger (oil-air cooler)

Fluid coming from the system flows back into the system through a tube cooled by means of a fan wheel.

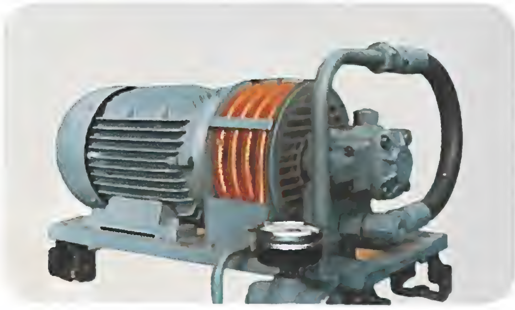
A basic advantage of the oil-air cooler is that the cool air required is available practically everywhere. The fan wheel must, of course, be driven in some way, and the cooler noise can not always be reduced.

The following picture shows one design:

Symbol

Accessories

This oil-air cooler is designed to function simultaneously as a coupling protection. The hub of the fan wheel is fixed to the motor shaft. Air flows from inside over the finned tube, which is wound several times round the fan wheel. The fluid flows back to tank through this ribbed tube and dissipates heat.

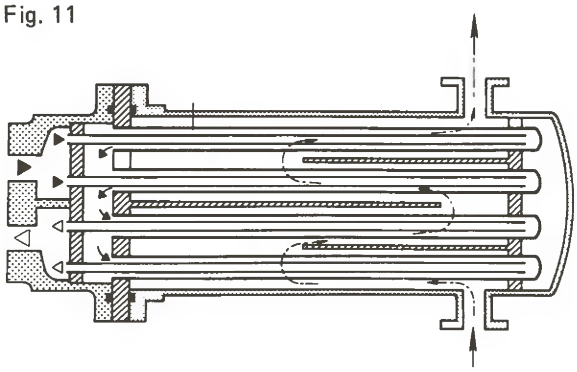


Motor/pump Group

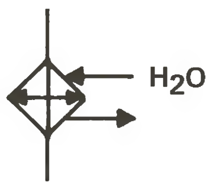
On the motor/pump group shown in the photograph, the leakage oil from the variable displacement vane pump is cooled before returning to tank by this cooler designed as coupling protection.

– Water Cooled Heat Exchanger (oil-water cooler)

Fig. 11



Symbol



These coolers feed either the water or the pressure fluid into cooling tubes, while the fluid or water circumcirculates the tubes.

Oil-water coolers have a greater cooling power than oil-air coolers, because the difference in temperature between the coolant and the pressure fluid is generally greater.

The place of application is very important, as a supply of coolant is required.

Heaters

Heaters are used to heat the pressure fluid to operating temperature. The fluid is heated by means of electrical immersible heating rods. Care must be taken, that the heat generation per unit area is not too great. This would result in overheating and carbonisation of the fluid on the upper side of the heating rod.

The maximum power with mineral oils should therefore not exceed 2 Watts per cm^2.
(Phosphate ester and water glycol,
0.6 – 0.7 Watt/cm^2)

Thermostats, Thermometers

Thermostats or thermometers are often used to check the operating temperature (sometimes along with a cooling or heating element). They are fitted in the oil tank. Contact thermometers or thermostats, which switch according to the cooling or heating system, are often used to keep a desired temperature constant.

Float Switches

A float switch serves to monitor the fluid level in a tank. It can monitor the maximum fluid level, the minimum fluid level or both.

If the level exceeds or falls below the level given, the float switch releases a contact by moving over the switching point set on a measuring rod. This signal is then either transmitted to a control unit or triggers a function (e.g. switches off the machine if the oil level is too low).

Notes

Methods of Connection

Methods of connection for hydraulic units

The methods of connection are as follows:

pipe connections, hose connections, cutting ring joints, clamp collar joints, screw joint with o-ring or metal seal on face, flange connections or quick lock couplings.

However, we shall describe more fully some other possibilities:

Subplate:

The subplate is the most frequently used method of connection.

Subplate

The valve side of the subplate is ground and has a porting pattern corresponding to the size and type of component (arrangement of and distance between the connection bores).

The connection surface of the valve is also ground, is fitted with o-rings and shows the same porting pattern. Most porting patterns are standard.

The valve is screwed on to the subplate. O-ring seals are used.

The threaded connections for the lines are on the underside of the subplate. The valve can therefore be removed quickly and easily without removing the lines. Where individual subplates are used, the amount of piping is comparable with line connections by means of threads in the housing.

Manifold Plate

Manifold plate for fitting several similar valves

The amount of piping and space required are reduced if manifold base plates can be used, in which the common supply and drain lines, also the inter-valve connections can be fitted. Apart from the supply and drain lines, only the service ports need to be piped.

Control Block

Control block for a plastic injection moulding machine to control all standard functions, with built-in and screwed-on valves.

Methods of Connection

The line connections of a control are in a control block. The valves can be fitted to the block, as for the subplate.

There are basically 3 types of block:

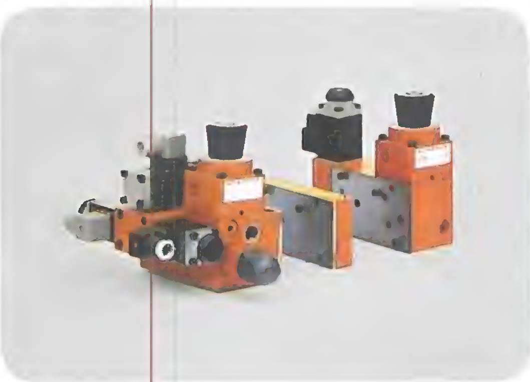
a) Line connection only, therefore no wear.

b) Line connection, anti-wear housing. Cartridge or screw-in valves are also used here (e.g. pressure control valve DBD or check valve), for which the control block represents the housing.

c) Line connection and at the same time "real housing". The control block is designed in such a way, that it is not only for line connections or as a mounting surface, but also forms the housing, for example, for a directional control valve spool.

These control blocks are generally designed for a particular application, and therefore bring about corresponding design and manufacturing costs. However, they are a very compact form of control, with which the assembly costs on the machine can be reduced to a minimum.

Control Blocks with Integrated Circuit



Single slide control for a lathe

These control blocks are cut to size for certain functions. They are designed for movement and working processes, as are required again and again in different machines or industries. The photograph shows an example of such control blocks – the control for a machining slide with synchronous cylinder.

The functions:

Left-hand directional control valve –
cylinder forwards, reverse stop.

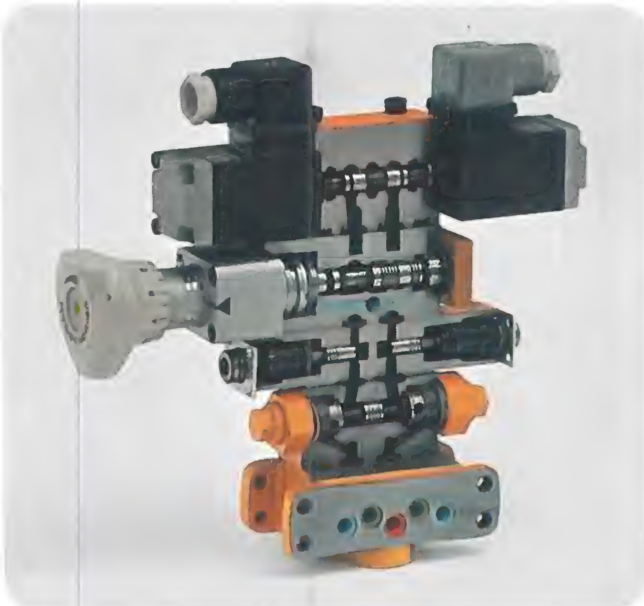
Upper directional control valve and flow controller –
fast forward, feed – on – off.

Pressure control valve on the front –
counterholding.

The block can be extended by the subplate shown on the right. This allows a second speed. As can be seen from the example, there are also models, where the functions can be extended using the "modular stacking" principle.

Stacking assemblies are for universal use, i.e. for fitting as many circuits as required with minimum piping.

Vertical Stacking



Vertical stacking assembly, built on a subplate for horizontal stacking

Following the principle of connections using a porting pattern, it is possible to form function groups compactly and without piping, by using so-called "sandwich plates". These have a mounting face with porting pattern and o-ring seals on both sides. They are placed between the subplate and primary valve, and fixed on by means of through bolts.
Individual or several plates for different functions can be used.

Sandwich plate for vertical stacking:
Check valve (type Z1S)
Double check valve (type Z2S)
Double throttle/check valve (type Z2FS)
Pressure relief valve (type ZDB, Z2DB with pressure relief in line A, B or both)

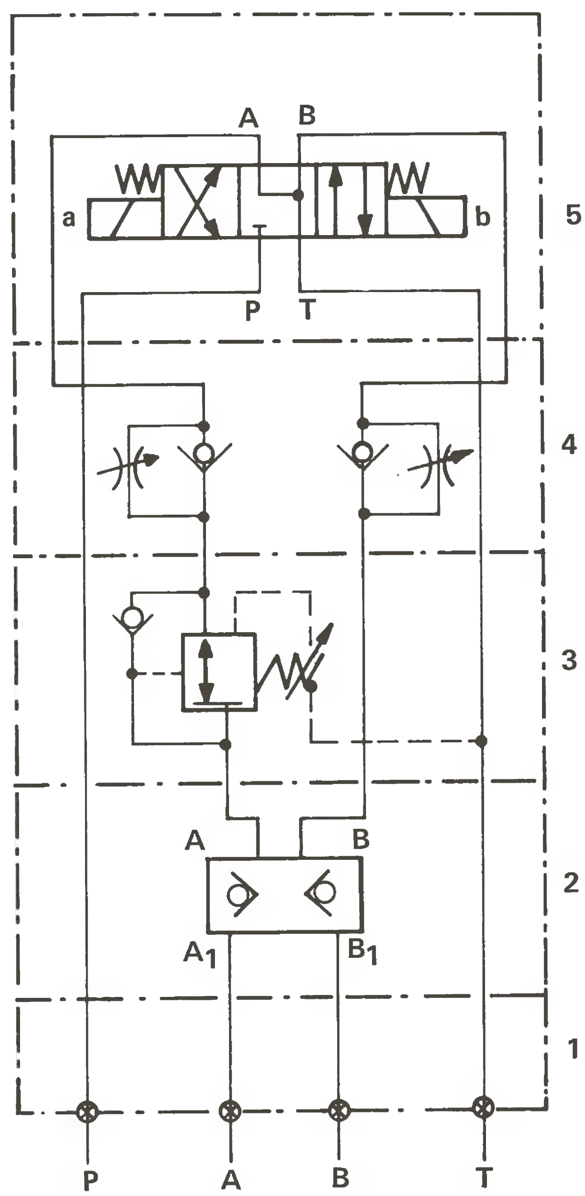
Methods of Connection

Pressure reducing valve (type ZDR with pressure reduction in line A or P)

The function of the valves has already been dealt with in the individual valve groups, so that we will consider the complete system here:

Circuit Diagram of a Vertical Stacking Assembly

Fig. 1



1 = subplate
2 = double check valve (Z2S)
3 = pressure reducing valve (ZDR)
4 = double throttle/check valve (Z2FS)
5 = directional control valve

By means of the subplate, the P line passes freely through the sandwich plate to the directional control valve. The T lines does likewise. If, for example, solenoid b on the directional control valve is switched, there is a connection from P to A.

The fluid therefore flows firstly through the double throttle/check valve 4 with throttle effect. Pressure is then reduced to the value set at valve 3. The last element before the subplate is the double check valve 2, for leakfree closure of the service connections A and B.

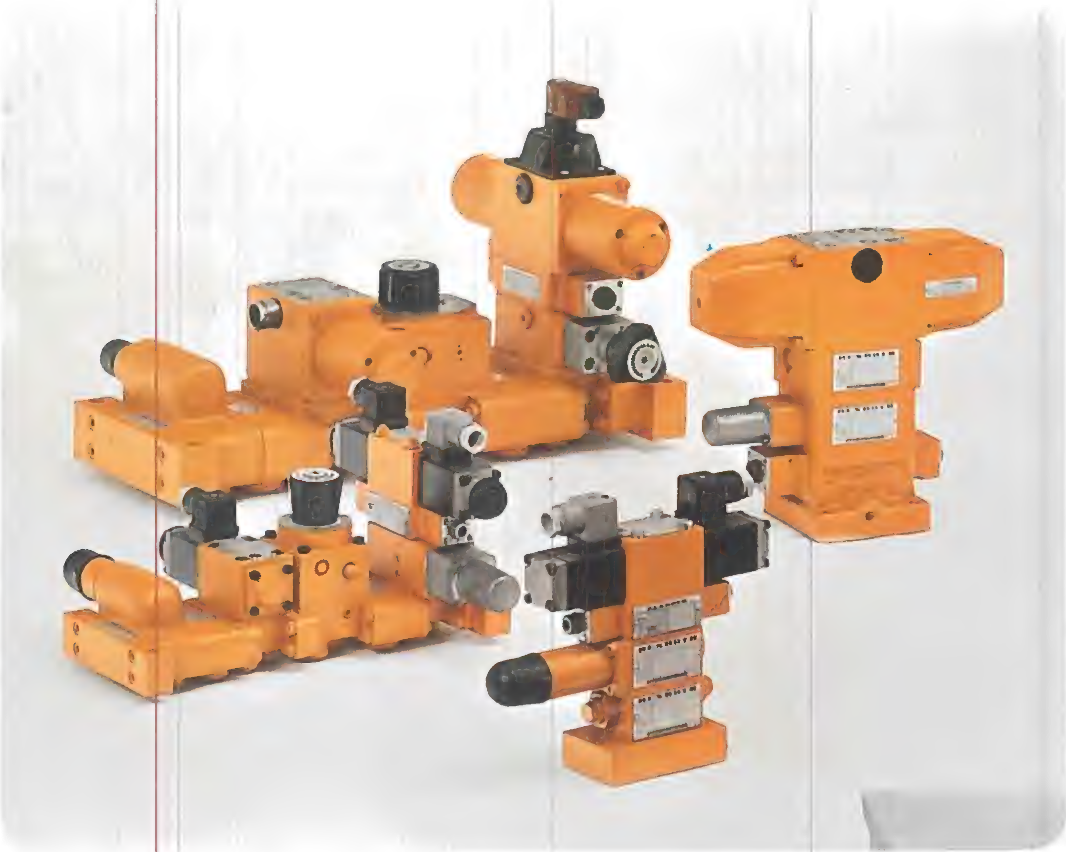
If the directional control valve is in neutral position, ports A and B of the check valve are unloaded. The user is hermetically closed.

The oil flowing back from the user via subplate port B can flow from B_1 to B (valve 2) on the check valve in valve 4, to directional control valve 5 and back to T.

When arranging the sandwich plates, please note that the check valve 2 must be the element nearest to the user. If an element with leakage (e.g. tolerance) is fitted in between, leakfree closure is no larger guaranteed.

Methods of Connection

Horizontal Stacking Assembly



left: horizontal stacking assembly sizes 6 and 10
right: vertical stacking assembly sizes 6 and 10

Compact controls with various circuits (fig. 2) can be formed by direct joining of specially designed stacking plates with built-on valves.

There are channels through the **stacking plates**. The individual plates are screwed together, and the channels thus joined.

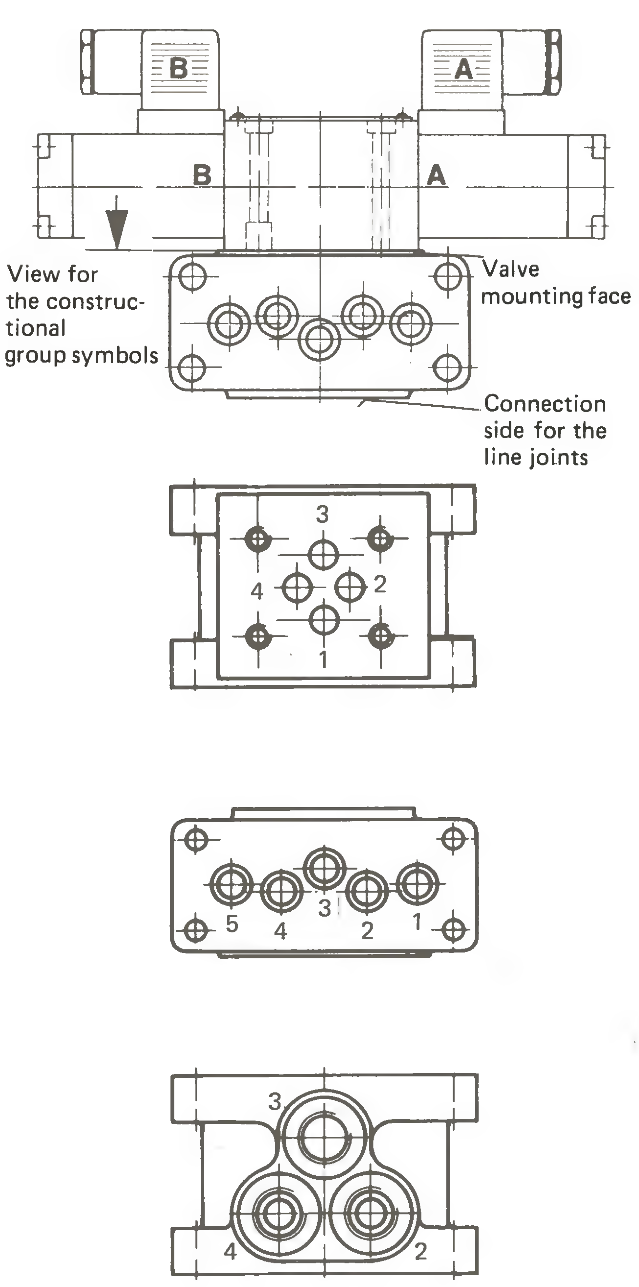
A **stacking subplate** has one sealed surface to the valve, as has the individual base plate. On the opposite side, threaded connections are available for certain lines, which can be used for joining pumps or can be plugged. Along with the subplates for directional control, check, poppet, pressure control and flow control valves, there are also **reversing plates**. They can be fitted between the subplates and allow any reverse line connection required for a circuit.

The line connections to the neighbouring stacking plates can be maintained or broken by fitting **dividing plates**. The dividing plate also serves to hold together the o-rings for sealing the stacking plates.

Either **fixing plates** or **reducing plates** are at the end of the horizontal stacking assembly. Fixing plates serve to close the three middle lines and have fixing holes for fitting the complete unit. Vertical stacking assemblies size 6 and size 10 or size 10 and size 16 can be connected using a reducing plate.

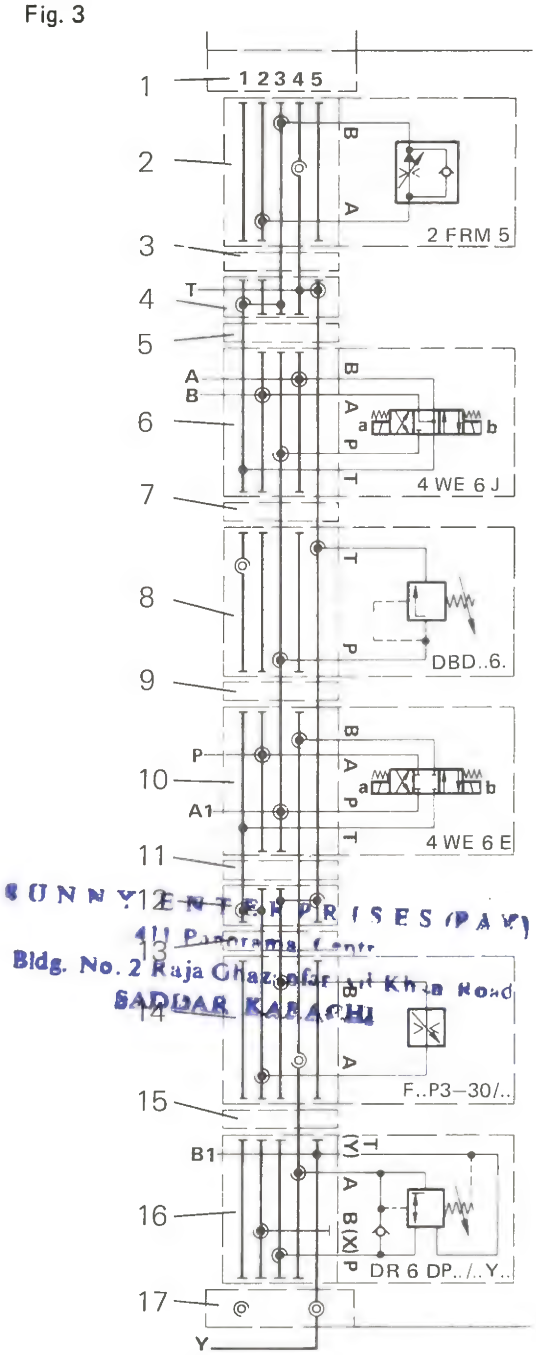
Methods of Connection

Fig. 2



Stacking Assembly Circuit with Plate Designation

Fig. 3



The diagram of a horizontal stacking assembly differs from the usual circuit diagram (fig. 3)

Example:

1 Fixing plate
2 Subplate for flow control valve
3 Dividing plate
4 Reversing plate
5 Dividing plate
6 Subplate for directional control valve
7 Dividing plate
8 Subplate for pressure relief valve
9 Dividing plate
10 Subplate for directional control valve
11 Dividing plate
12 Reversing plate
13 Dividing plate
14 Subplate for fine throttle
15 Dividing plate
16 Subplate for pressure reducing valve
17 Fixing plate

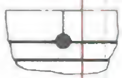
Methods of Connection

Symbols

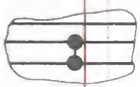
Main oil passages

Pilot oil passages

Drain oil passages



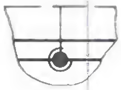
Connection porting in the valve mounting face of a subplate



Connection porting in an inter-connecting plate

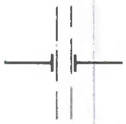
Plugged threaded port on line connection side

Open threaded port on line connection side



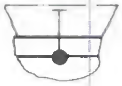
Connection porting on the valve mounting face and plugged threaded port on line connection side

Connection porting on the valve mounting face, and open threaded port on the line connection side



Passage between two adjoining subplates or inter-connecting plates blanked off using "blank" dividing plates

Passage between two adjoining subplates or inter-connecting plates continued using drilled dividing plate



Passage between subplate and valve blanked off

Notes

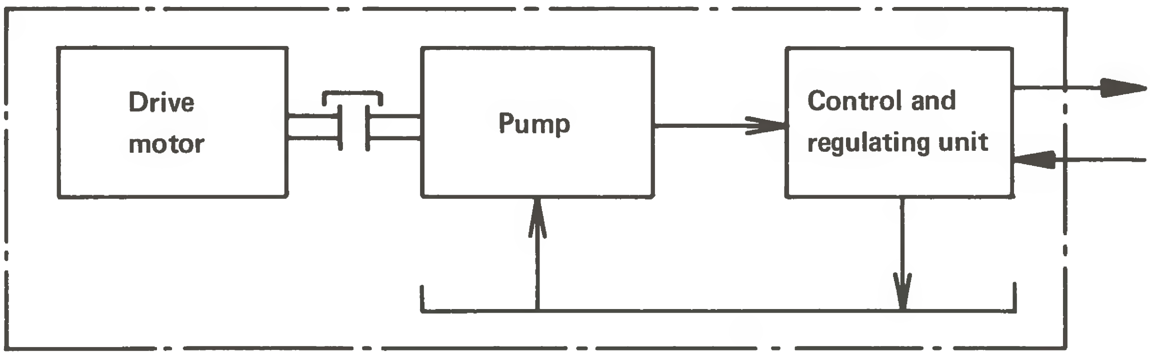
Programme Summary

- Hydraulic Power Units and Plant
- Small Power Units
  - Tank Capacity, 10, 20, 40 l
    - with motor/pump group
    - Control and Monitoring Elements
    - with connection blocks
    - with front panel
- Pump/Tank Units
  - with motor/pump group
  - with front panel in element design
  - with front panel in closed design
- Motor/Pump Groups
- Motor/Pump Stations
- Rectangular Oil Tanks
  - Tank capacity
    60 to 1 000 l
    to 10 000 l
- Round Oil Tanks (Tank Stations)
  - Tank capacity
    1 000 – 10 000 l (standard)
    and larger
- Cabinet Units
- Valve Stands
- Valve Tables
- Accumulator Stands
- Training and Test Units
- Special Units

Hydraulic Power Units

Hydraulic Units

Fig. 1



A hydraulic power unit refers to the grouping together of individual components into one unit (fig. 1).

The grouping and constuction will vary according to the application. In spite of this, units can be built to certain standards and instructions. In our case, these are the power unit standards (AB standards), which result from our years of experience in the building of hydraulic units. These refer to small standard units as well as to standard units, motor/pump groups, control stands, tank stations, accumulator stations, special power units or other units.

First of all, we shall deal with the basic elements of a power unit and their main purposes.

Tanks

The tank has a number of very different functions.

– Intake of the oil supply.
  If possible, the tank should be able to take the total oil supply of a system. The varying volume, dependent on the users and the working cycle, must be taken into consideration. Other leakages are replaced from the oil reserve in the tank.

– Cooling (heat dissipation)
  Each transfer of energy causes losses, which, in hydraulics, are given off as heat to the hydraulic fluid. The degree of this loss determines the degree of efficiency. The total loss on hydraulic units is made up of the power losses in the lines, at pumps, motors and valves (internal leakages), throttle losses and the energy transfer at pressure control valves.

The relevant amount of heat is transmitted into the oil. The pipes, control elements and tank dissipate a part of this heat to the surroundings. The remainder heats up the oil and unit parts, until there is a state of equilibrium between the power loss supplied and the amount of heat dissipated. The temperature set then is the inertia temperature. In order to operate without separate cooling, the inertia temperature must be equal to or less than the maximum operating temperature.

The power loss, which is of interest to us, is made up of:

$$p_{loss} = p_{loss\,pump} + p_{loss\,valves} + p_{loss\,motor}\ [kW]$$

Over a total grade of efficiency, p_{loss} can be roughly calculated:

$$p_{loss} = p_{total} \cdot (1 - 0.7 \text{ to } 0.75)\ [kW]$$

Heat, which is dissipated by the tank:

The amount of heat ($W_{A\,tank}$), dissipated by the tank depends on

tank size
amount of fluid in the tank
difference in temperature between inside and outside
place of installation.

The following is generally accepted:

$$W_{A\,tank} = \Delta T \cdot A \cdot k\ [kcal/h]$$

Hydraulic Power Units

Temperature difference ΔT in °C
Dissipating tank surface A in m²
Heat coefficient k [kcal/m² · h · °C]
or W/m² · °C; 1 kWh = 860 kcal.

The following values can be stated for k:

k ≈ 5 with bad air circulation, heat peak, unfavourable installation

k ≈ 10 installation in normal halls, normal air circulation on all sides

k – 20 with high degree of air movement, e.g. unnaturally caused air movement.

If the heat is dissipated only by means of the tank, the set temperature difference between oil and air results as:

$$\Delta T = \frac{p_{\text{total loss}} \cdot 860}{A \cdot k} \; [{}^{\circ}C]$$

Fig. 2: example of heat dissipation for Rexroth standard tanks.

The length of operation naturally influences the amount of heat which occurs. For calculating ΔT, p_{loss} was considered as a constantly occuring heat.

– Elimination of air

Mineral oils contain broken-down air. The air breakdown capacity is related to the pressure and the temperature. Air in the form of little bubbles can therefore come away from the oil in a system. This air must be eliminated in the tank. The upper oil surface should therefore be as large as possible

– Elimination of Contamination

In spite of adequate filtration, dirt particles, slurry and deposits caused by age can accumulate in the system with increasing length of operation.
These particles should be deposited on the tank base. The design and arrangement of the suction and return lines are therefore very important.

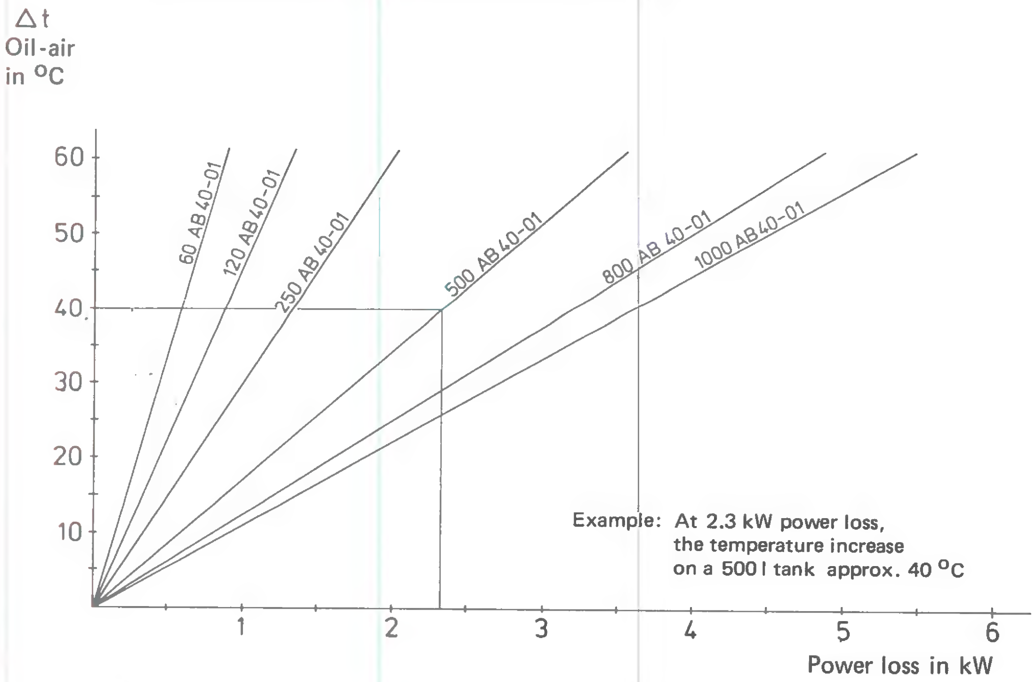
The end of the suction and return lines should be cut at an angle of 45°, and be positioned in such a way that they do not influence each other (slopes lying away from each other). With larger tanks (over 1000 litre capacity) or very strong oil movements, baffles are fitted. They cause a division between the suction and return line regions.

– Fitting of Motor/Pump Groups and Front Panels

On standard power units, it is usual to fit the motor/pump group and perhaps also the front panel with the control elements on the tank. This must be taken into account when designing the upper side of the tank.

Fig. 2

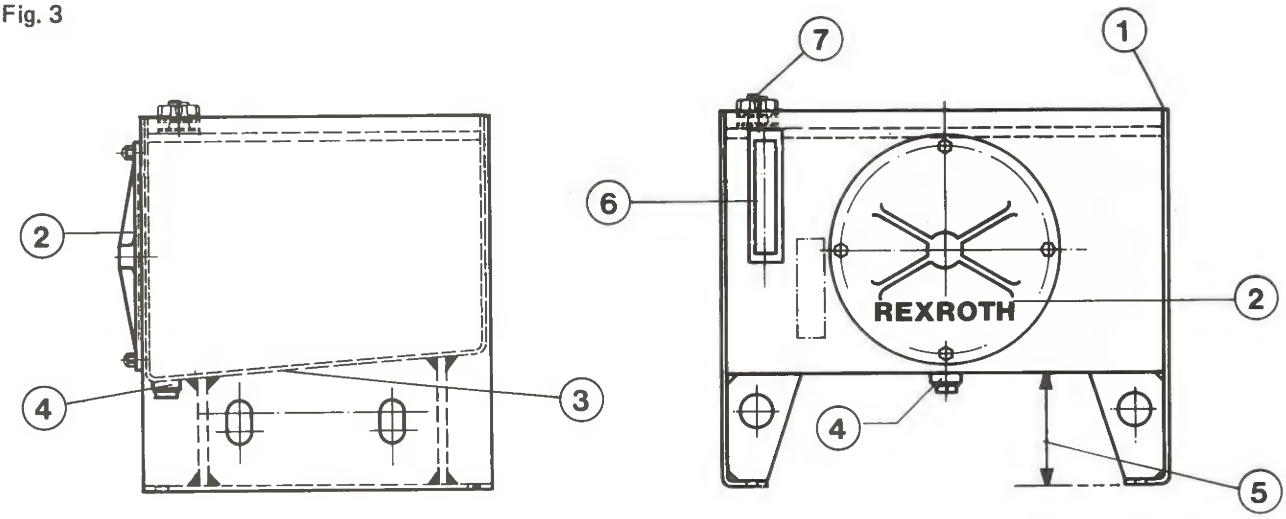
Heat dissipation of energy loss on tanks to AB 40-01



Hydraulic Power Units

Tank Design (fig. 3)

Fig. 3



There are various requirements, which a tank must meet

- Distortion-free model (fitting of other elements)
- Waste oil tray 1, so that the oil is caught when the built-on elements are changed
- Cleaning cover 2
  The opening must be large enough for all surfaces and corners in the tank to be easily accessible.
- Suitable tank base 3 and drain plug 4
  The tank should have a drain plug at the lowest part of the base, so that it can easily be emptied and flushed.
- Ground Clearance 5
  A tank over 40 litre capacity should be 150 mm clear of the ground, for better cooling, cleaning and transporting possibilities. The clearance also ensures good air circulation. The base surface can then also be used as a dissipation surface for heat dissipation.
- Oil Level Indicator 6
  The maximum and minimum oil levels can be read off here from outside. If the indicator is long, the level at any time can be easily seen.
- Filler/Breather 7
  An air filter should be provided for the necessary bleeding of the tank and simultaneous cleaning of the sucked-in air.

Please note that the rate of air flow must be greater than the maximum varying volume of the tank. If the design is too small, this could lead under certain circumstances to negative or excess pressure in the tank. This must be avoided. A filter mesh should also be provided in the filler opening.

- Tank Size
  The size of the tank depends on:
  pump flow (generally 3 to 5 · Q)
  air cushion (10 ... 15%)
  equipment to be fitted
  cooling (see cooling)
  volume of the unit

Hydraulic Power Units

Types of Power Unit

Small Power Units

type U1 - 10 S 10/...

These small power units with a tank capacity of 10 or 20 litres comprise (fig. 4):

- alluminium alloy tank with cooling fins, which also strengthen the unit
- tank cover, pump support
- el. motor, coupling, pump
- return line filter
- pressure control unit
- oil level indicator
- oil drain screw
- air filter

(The purpose and function of the individual elements were described in the chapter "Accessories").

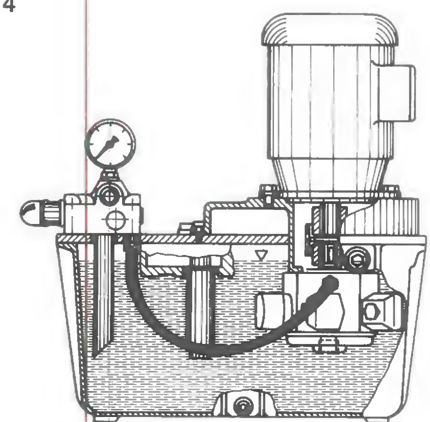
The el. motor is fitted vertically (model V1). The pump is suspended in oil.
In spite of this compact design, it is still possible to fit a control. Either a vertical or horizontal stacking assembly can be fitted using a connection block. The block has a corresponding connecting surface and comprises pressure relief valve, return line filter element and pressure gauge connection with shut-off valve.

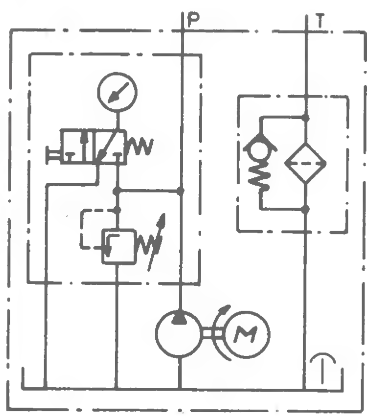
As an example, a power unit with connection block and vertical stacking assembly:

Small power unit type U2 - 20 S 10/...

The circuit diagram is shown on page 5.

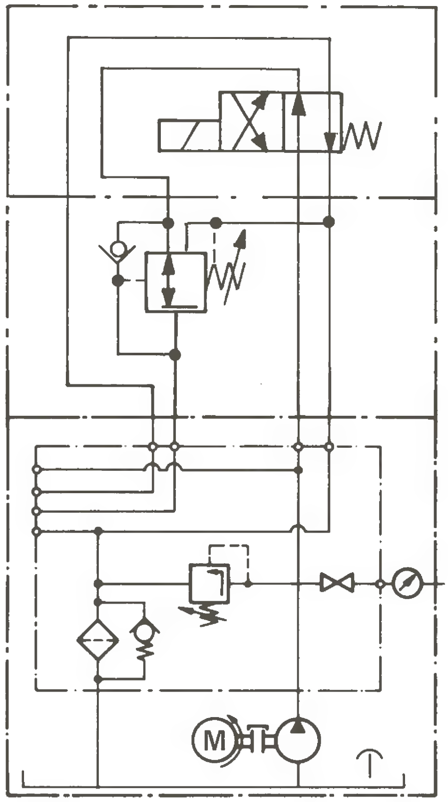
Fig. 4





Hydraulic Power Units

Fig. 5



Important technical data:

tank capacity	10 l	20 l
oil drain	... 4 l	... 8 l
max. power	1.5 kW	4 kW

Small Power Unit with 40 litre tank

Small power unit type U3 - 40 SF 10/...

The basic design corresponds to the small power units with 10 or 20 litre tank.

The el. motor with the pump can also be fitted horizontally. If the controls to be fitted cannot be carried out with vertical or horizontal stacking assemblies, it is also possible to fit a small front panel.

Tank feed can also be fitted, if desired.

Pump/Tank Units

The basic element of the pump/tank unit is the rectangular standard tank size 60 to 1000 (60 to 1000 litre tank capacity).

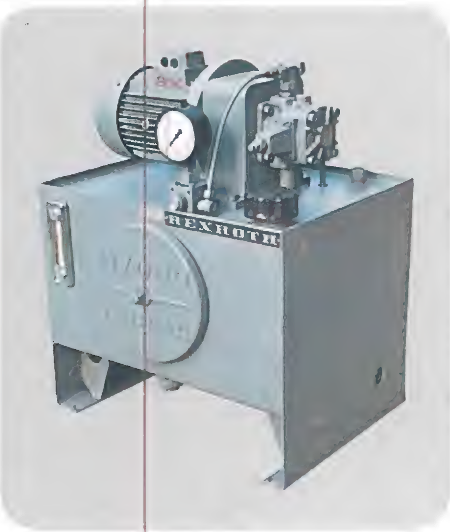
The motor/pump group can be fitted in 3 different models:

Model V1: the el. motor is arranged vertically on the tank cover. The pump is immersed in oil.

Model V1

Hydraulic Power Units

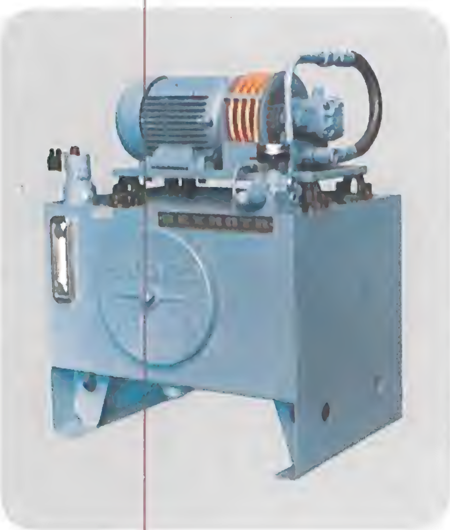
Model B5: Pump and motor have a mounting flange. It is best to mount them on a foot mounted bell-housing, which simultaneously forms the coupling protection and saves aligning the shafts to one another.



Model B5

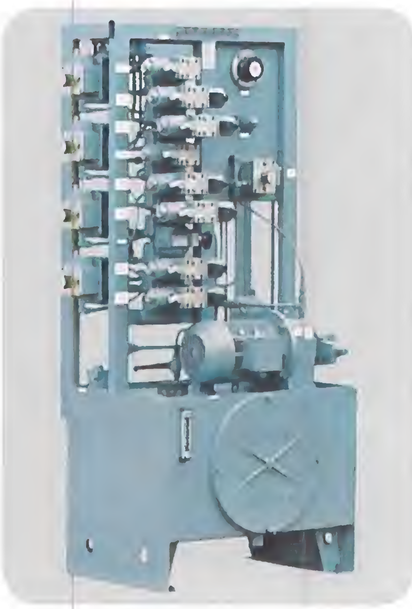
Model B3: Pump and motor are foot mounted. They are fitted on a base frame and to the tank by means of swinging metal elements. Sound insulation is thus also achieved. The pressure line is carried further by means of a hose line.

Also, the lines through the tank cover (suction line, return line) have no metal connection to the cover. Flexible seals are used for the pipe conduction.



Model B3

Front Panel in Element Design



Pump/tank unit with front panel in element design

The valves are fitted on individual elements (plates) which are then screwed on to a rectangular frame from the back. The design has several advantages.

- easy frame production
- stock holding of the individual elements
- can be combined as desired
- alterations later are easily possible (without flame cutting)
- air noise deflection is greatly reduced.

The parts in the stacking plates remain easily accessible after stacking assemblies have been fitted.

(see photograph on top of next page)

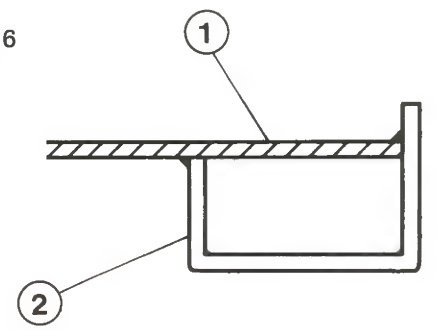
Hydraulic Power Units

Front Panel in closed design

Hydraulic power unit AHAG heavy duty front panel design

A piece of smooth metal 1 is strengthened by welding on a C piece 2 (fig. 6).

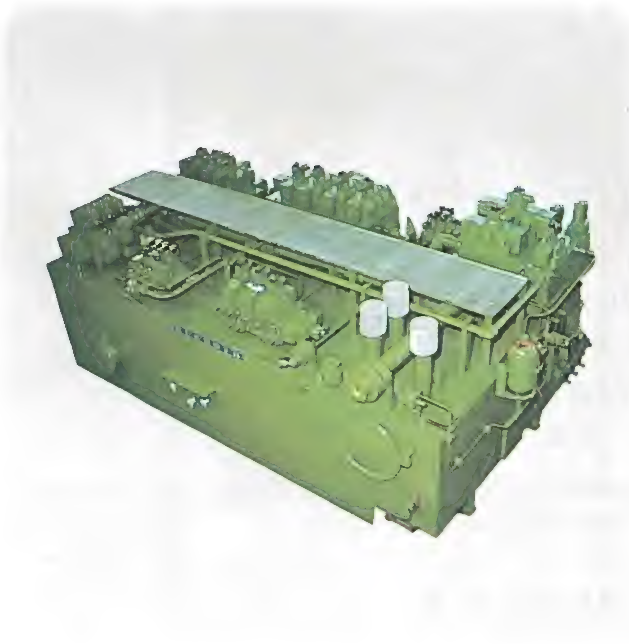
Fig. 6



This model is used for equipment over size 22.

Rectangular Oil Tanks, Circular Oil Tanks

Strengthened rectangular or circular tanks are used for larger volumes, i.e. over 1000 litres capacity. Where rectangular tanks are used, it is possible to fit control elements or motor/pump groups direct.



Complete drive station for a 2500 t aluminium extrusion press

The basic tank model includes:

- cleaning cover
- oil drain tap
- oil level indicator
- filler/breather

Hydraulic Power Units

Fig. 7

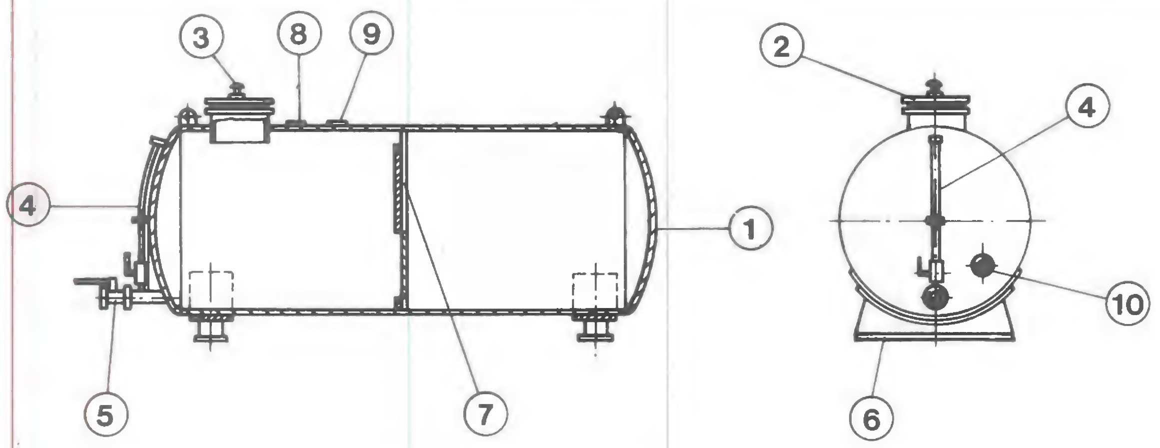


Fig. 7 shows a section of a circular tank

1 Circular tank
2 Cleaning cover
3 Filler/breather
4 Oil level indicator
5 Oil drain tap
6 Tank foot
7 Optional dividing wall (with cleaning hole)
8 Optional leakage port
9 Optional thermometer, thermostat
10 Heating

A stopcock should be provided in the suction line, to separate the oil supply from the tank to the drive station. We recommend that the position of the stopcock (open or closed) should be monitored using a limit switch. Electrical locking prevents the pump being started when the suction tap is closed.

Tanks, motor/pump group and control are designed as separate units, for applications for circular tanks and in general, on large installations. Of course, there are exceptions to this, as shown by a few examples of special power units (see page 192).

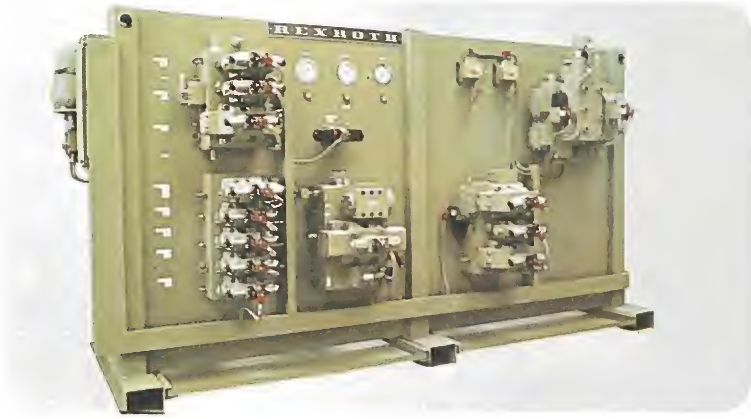
Motor/Pump Stations

Motor/pump station, type APS

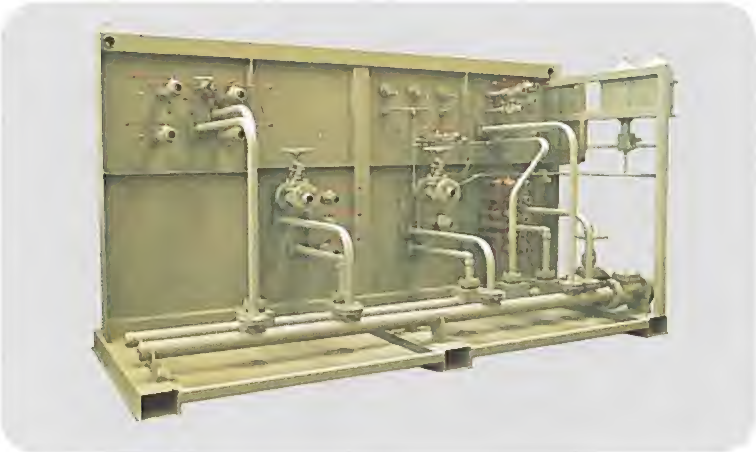
The motor/pump group is built on a heavy duty base frame. As shown in the photograph, it can also be fitted with flexible elements for noise insulation. The coupling protection is to DIN 31 001 safety instructions. An additional oil tray and a small front panel have been fitted on the APS shown.

Hydraulic Power Units

Valve Stands



Valve stand (front), type AVS



Valve stand (rear)

The construction of a valve stand is similar to that of the front panels. Equipment is fitted on the front, the pipes are fed to the users or into multiple lines on the back.

Valve Tables

Valve table, type AVST

Valve tables have proved to be a very good method for fitting heavy duty control plates. The control blocks or control plates are mounted vertically and thus easy access to the valves is guaranteed.

Cabinet units (type ASAG) are included in the range of standard units. In this case, the control fitted on the tank is mounted in a cabinet. This gives a neat construction and also makes noise reduction possible.

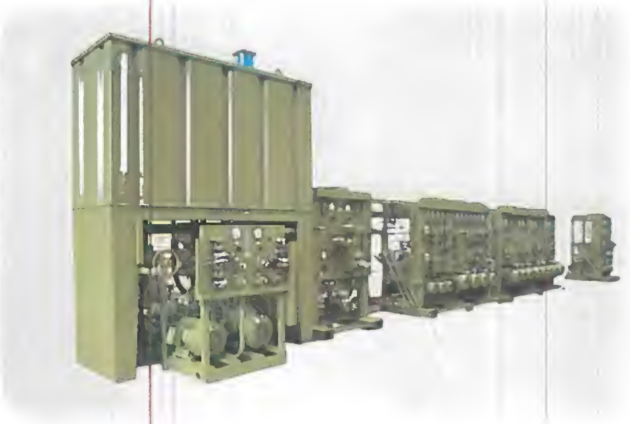
Accumulator Stand type ASS
for Pressure Accumulators sizes 32 and 50

The accumulators are fixed on a frame in one or two rows, according to the number of accumulators.

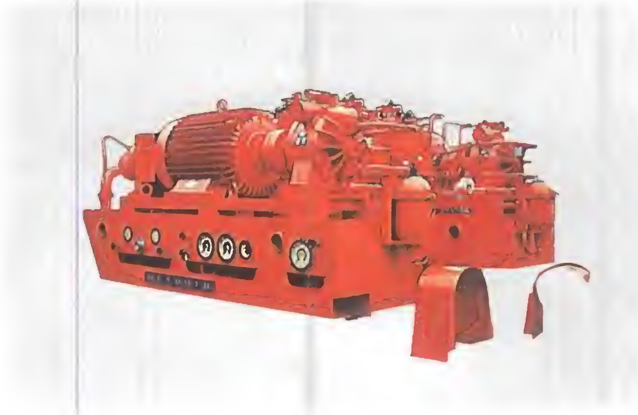
Hydraulic Power Units

Special Units

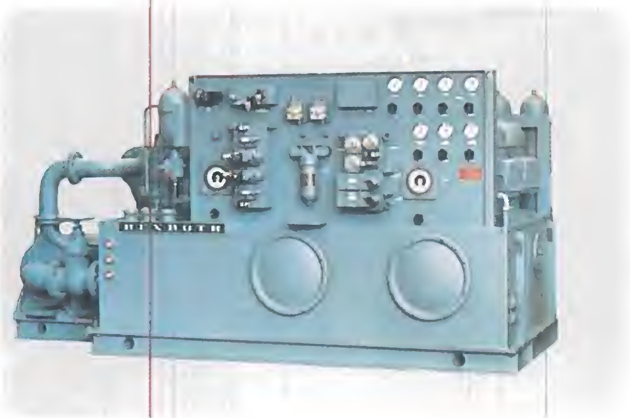
Finally, we shall show you a few special units, which have been designed for special machine and space conditions.



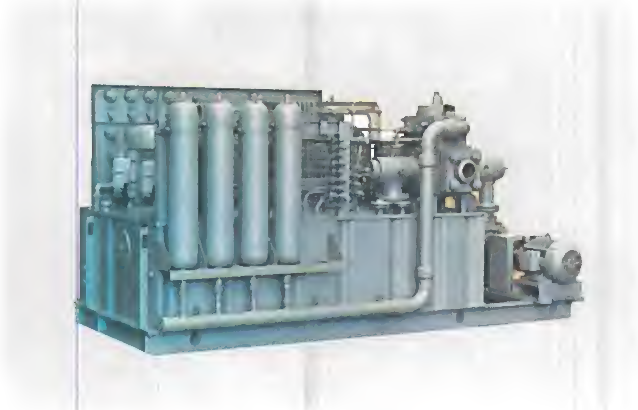
Drive station for Stör floodgate control



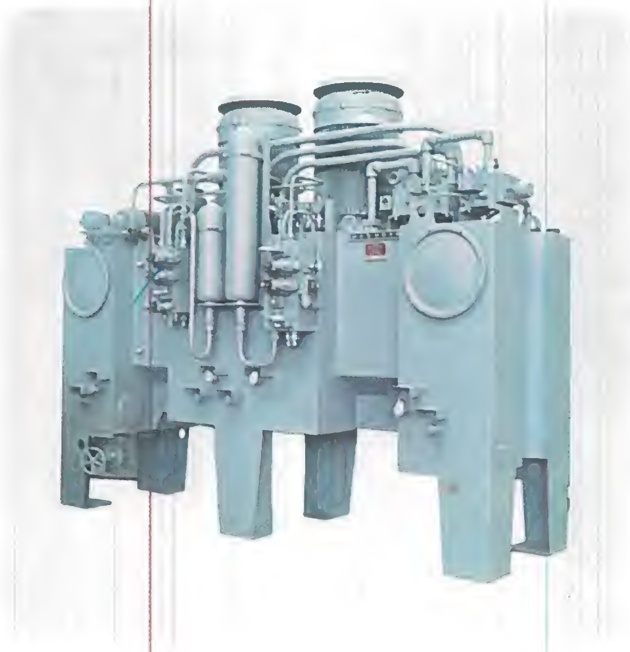
Complete drive station for a forging manipulator



Drive unit for a spindle press



Rear view of picture on left



Drive unit for a 400 t drawing press

Notes

Hydraulic Components in the Industry Sectors

Apart from the standard oil hydraulic elements already described, there are other components specially designed for the various industry sectors.

It is not possible to deal with these individually.

We wish to summarise on the following pages the elements and systems which are standard for the various sectors of industry.

We shall also mention the valve series for operation with:

oil/water emulsion

water

water glycol

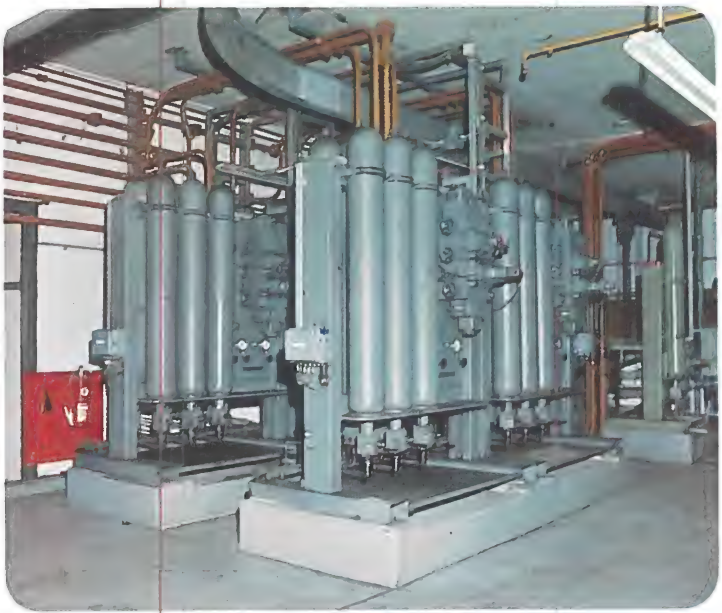
phosphate ester

mineral oil

The valve design differs from that of the oil hydraulic elements, as the special characteristics of water must be taken into consideration (e.g. viscosity, lubrication, temperature, corrosion, etc.).

This refers mainly to the design of the function parts and the materials.

Hydraulics for the Steelworks Industries



Accumulator station to supply a
Cowper and blast furnace plant

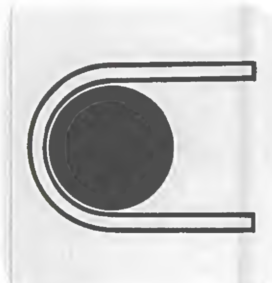
View of control stands with the valve for
Cowper and blast furnace armature controls

Throttle valves for subplate mounting (up to size 82)
for flange connections (up to size 102)

Directional spool valves
for subplate mounting (up to size 82),
for flange connections (up to size 102)

Pilot operated check valves
for subplate mounting (up to size 82),
for flange connections (up to size 150)

Variable displacement pump A2V with
pressure controller



Hydraulics for Machine Tools

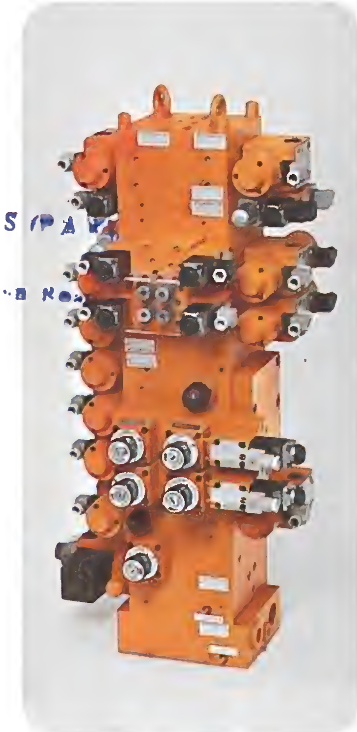
Variable displacement vane pump type V3

Tracer valve

Single slide control

Multi-flow controller

Mechanical unit
for table reversal



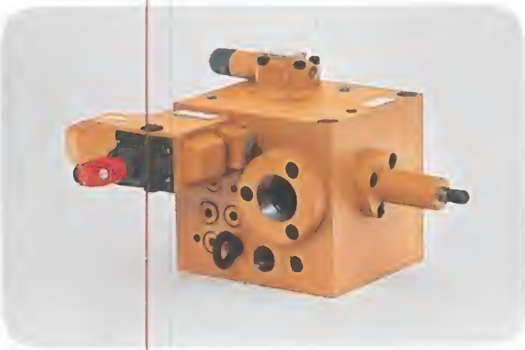
Tower stacking assembly

Transverse slide control

Control for
chucking device

Variable displacement
vane pump type V4

Hydraulics for Presses



Small press modules with
safety circuit

Variable displacement pump A2V
with HP control, direct operated

Variable
displacement pump
A2V with HP control,
pilot operated

Directional spool valves for subplate mounting (up to size 82)
for flange connections (up to size 102)

Pilot operated prefill valves,
sizes 100 – 350

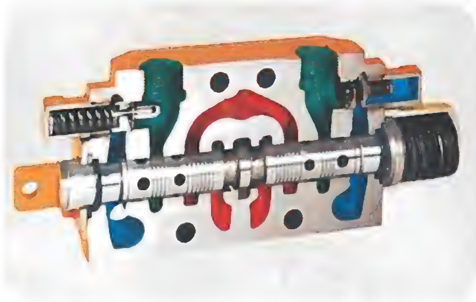
Senk modules sizes 25 – 100

Pilot operated prefill valves,
sizes 40 – 80

Mechanically operated
3 directional regulating valve

Hydraulics for Mobile Machinery

Slow speed fixed
displacement motor
with hydraulic brake



Directional control element of a
sandwich control block (sizes 10 – 22)

Pressure switch
for overload warning

Mono control block (sizes 16 – 32)

Pilot oil unit

Variable displacement motor A8V
with summation HP control

Pilot unit

Rotary transmission

Variable displacement motor A6V
with electrical control

Variable displacement
motor A6V
with automatic control,
high pressure related

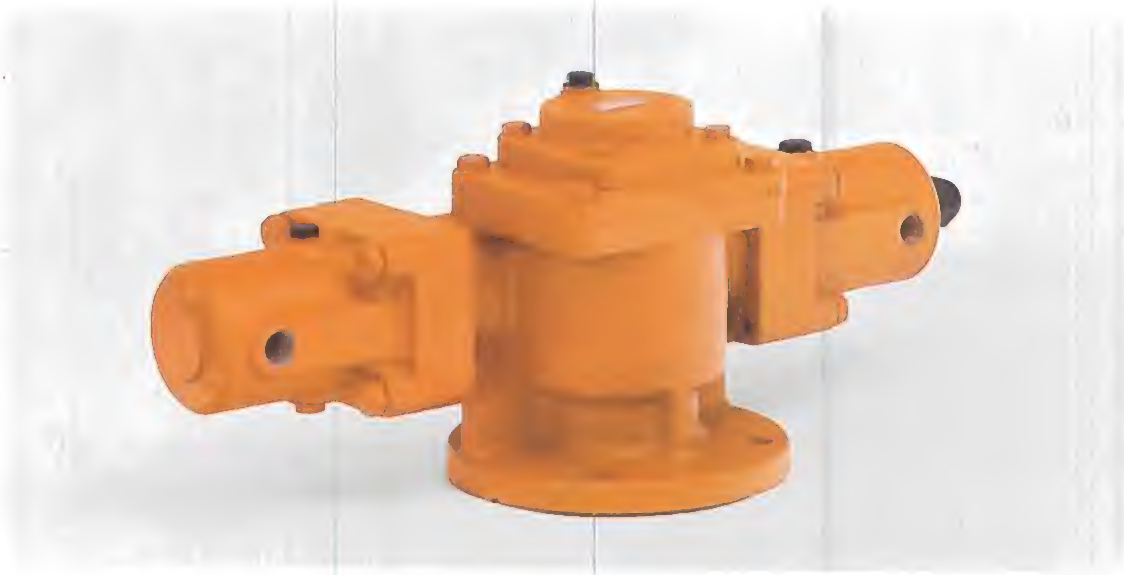
Hydraulics for Marine Applications

Hydraulic cylinder CD 250/CD 350,
swinging eye with spherical bearing at the cylinder cap
and spherical eye at the cylinder head

Directional spool valve,
electrically operated,
weatherproof (insulation: (Ex) sG4),
sea water resistant

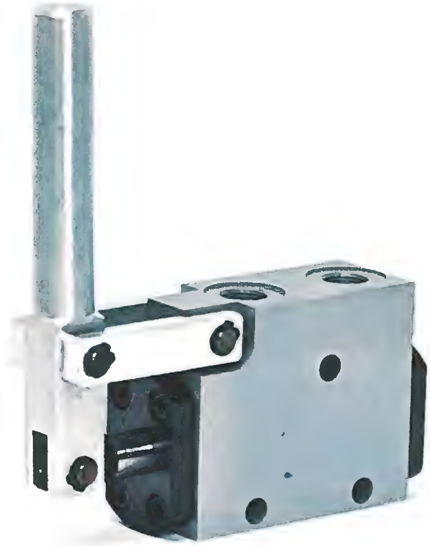
Directional spool valve,
manually operated, sea water resistant

Directional spool valve,
electro-hydraulic operation,
intrinsically safe (insulation: (Ex) isG5),
sea water resistant



Actuator type B with hollow shaft

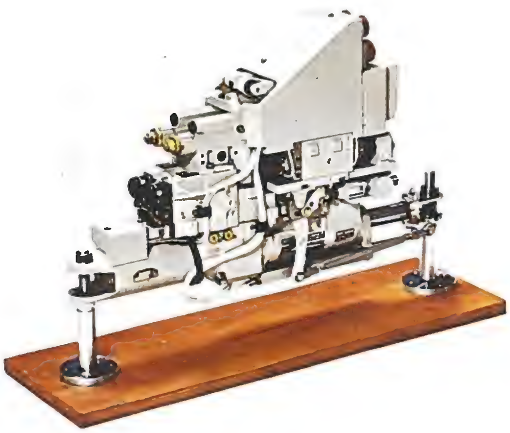
Hydraulics for Special Applications



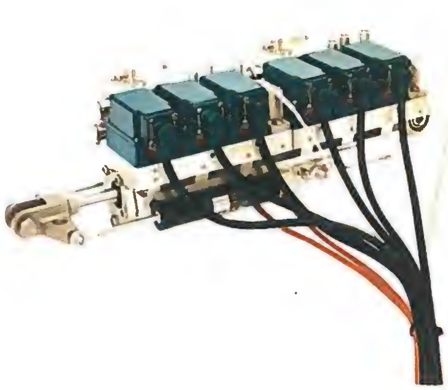
Double acting hand pump, p up to 315 bar
V = 25 cm^3/double stroke, m = 2.3 kg

Control unit, vertical stacking design,
to operate a coupling cylinder

Winch control for senser carrier
on a sea research buoy



Side rudder adjustment unit for MRCA (Tornado)



Double-triplex-majority system

Circuit Diagrams

The hydraulic circuit

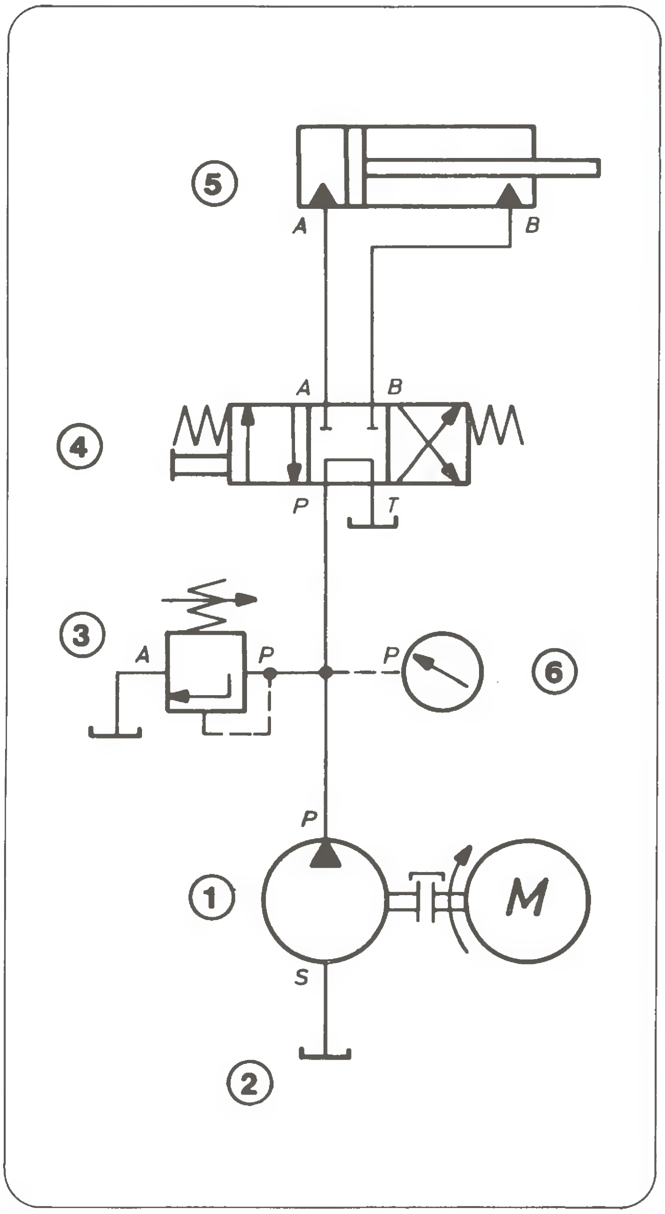
A hydraulic circuit diagram shows how a hydraulic circuit is built up.

The individual pieces of equipment are shown as symbols and connected to one another accordingly. Pipe connections are drawn as lines.

The sequence of functions of a hydraulic system can be seen from the circuit diagram. In the case of extensive diagrams, a work sequence diagram is usually also available, so that the exact timing of the work sequence can be seen.

If you look at a larger number of circuits, you will notice that different switching sequences occur again and again as parts of extensive hydraulic systems.

We shall now describe in detail a few basic examples of circuit.



Simple Hydraulic System (open circuit)

Here we have a hydraulic system in its most simple form. A pump 1 with fixed flow sucks fluid from a tank and feeds it into the system connected to it. In zero position of the manually operated directional control valve, the hydraulic fluid circulates almost without pressure from the pump to tank 2.
The valve is spring centered.

When the directional control valve 4 is operated into its left switching position (parallel arrows), fluid reaches the piston chamber of cylinder 5. The piston rod travels outwards.

The speed of outward travel depends on the pump flow and the cylinder size (piston area).

The force available at the piston rod is dependent on the piston area and the maximum system pressure. The maximum system pressure and thus the loading of the hydraulic system is set at the pressure relief valve 3.

The actual pressure available, determined by the resistance to be overcome at the user, can be read off at the pressure gauge 6.

Circuit Diagrams

Hydraulic Systems with Directional Control Valves in Series Switching Sequence.

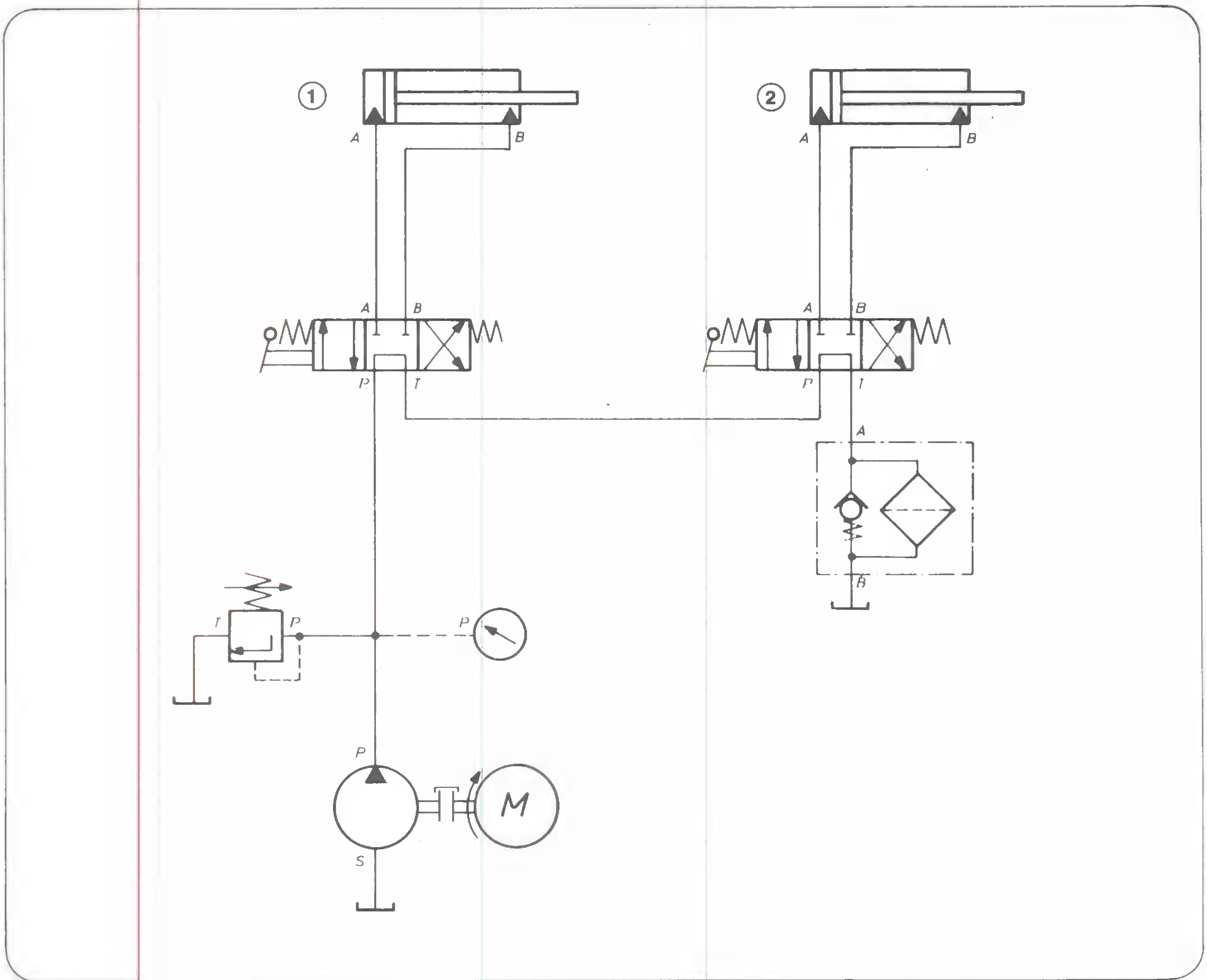
If you add to the previous system by bringing the return line from the first directional valve through one more or several valves, as shown, then you have a so-called "series switching sequence".

However, when using this arrangement, please note that several users cannot be operated at the same time, without influence related to force and speed.

The following conditions result:

Pressure according to the stroke force required and the piston area is necessary to operate the cylinder 2. This pressure affects the annulus area of cylinder 1. The system pressure finally required at cylinder 1 results from the external force affecting the piston rod and the pressure force, resulting from the pressure required by cylinder 2 and the piston annulus area of cylinder 1. If the pressure force available at cylinder 1 is greater than the sum of the individual forces, both cylinders travel outwards. The speeds from cylinder 1 to cylinder 2 are in the same ratio as the piston area of cylinder 2 to the annulus area of cylinder 1.

Before reaching the tank, the fluid flows through a return line filter.



Circuit Diagrams

Hydraulic System with Several Directional Control Valves in Parallel Switching Sequence

A variable displacement pump 1, whose flow can be altered by means of a variable displacement motor 2, sucks in fluid and delivers this into the hydraulic system connected to it.

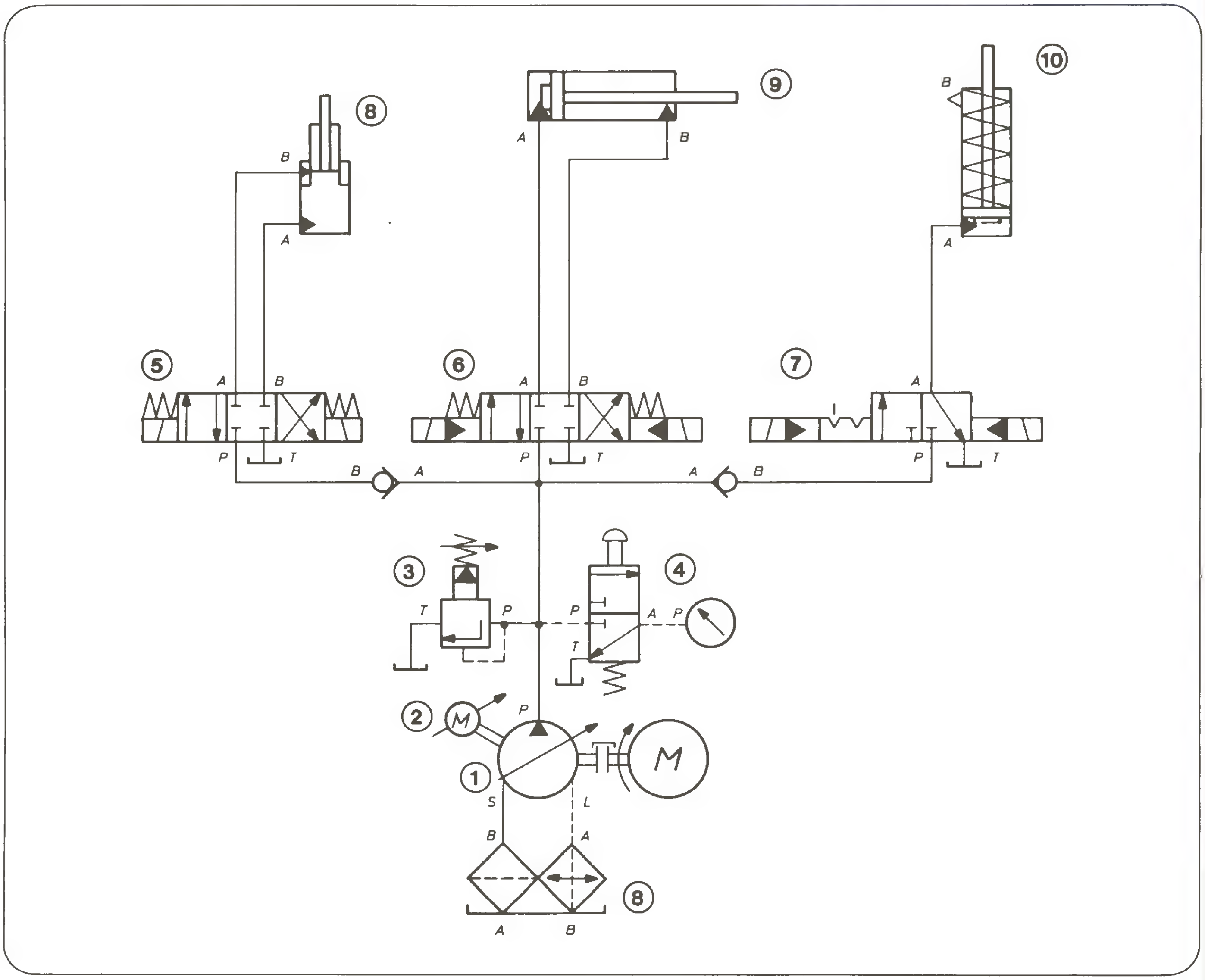
The line is divided and cylinders 8, 9 and 10 are supplied via directional control valves 5, 6 and 7. The directional control valves and the cylinders are parallel to each other. In our example, directional control valves 5 and 6 have ports P, A, B and T closed in centre position. Directional control valve 7 has port P closed in the right switching position.

The system pressure set at the pilot operated relief valve 3 is effective to the directional control valves. This can be read off at the pressure gauge after pushing the button on the 3/2 directional control valve 4.

A double acting telescopic cylinder 8, a differential cylinder 9 with constant end position cushioning on the piston side and a single acting cylinder with spring return 10 are shown as users.

The parallel switching sequence allows movement of several cylinders at the same time only when there is adequate fluid available to maintain the working pressure required. Otherwise, the pressure will always set itself at the value of the lowest resistance, i.e. at the lowest pressure value. This means that the cylinder with the lowest required pressure will travel out first.

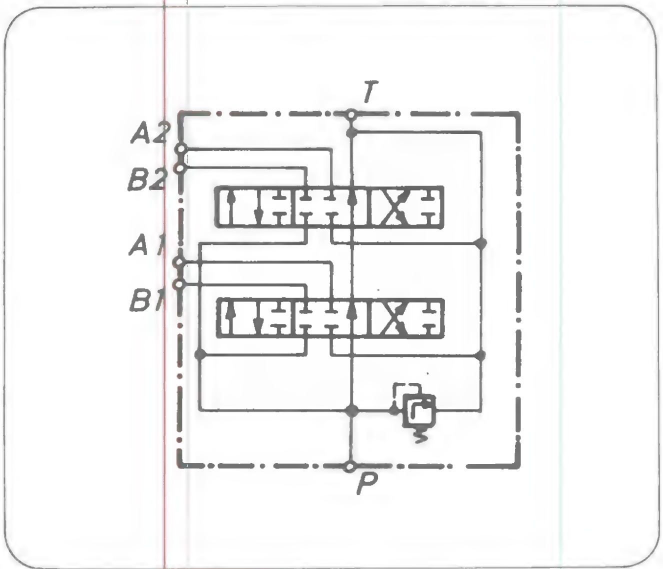
If the first cylinder is in end position, the pressure continues to rise until it reaches the value for the next cylinder The cylinders therefore travel out in relation to the load pressure required.



Circuit Diagrams

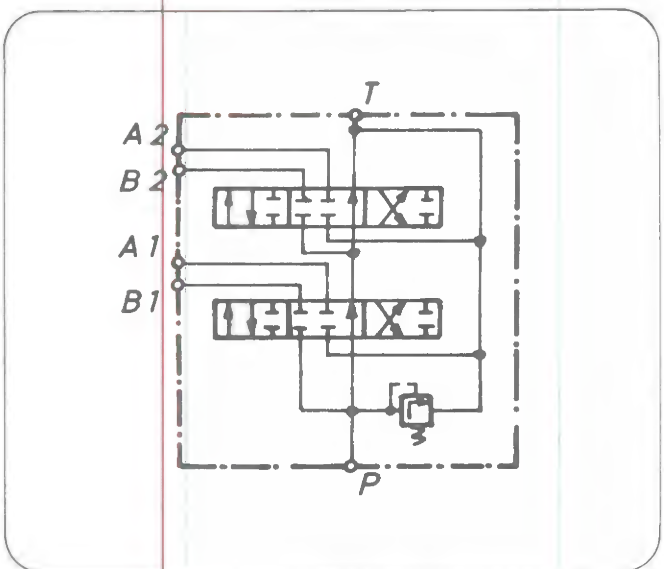
Directional Control Valves – Switching Circuits

The following switching circuits show examples of application in the mobile sector:



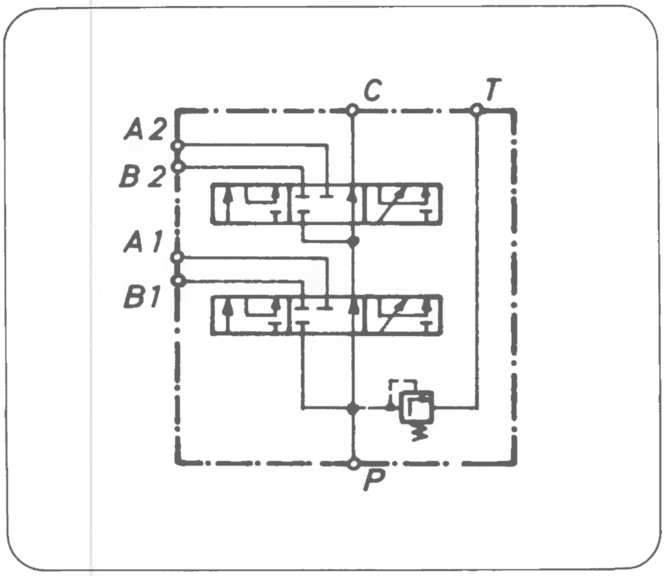
Parallel Switching Sequence of Valve Elements

Each valve is connected with the P line, all users can be controlled simultaneously. However, the oil is dispensed according to the resistances of the user circuit.



Tandem Switching Sequence

Oil supply only via bypass line, simultaneous control of several users not possible – priority – or also safety circuit.



Series Switching Sequence

Return oil is utilized, valve 2 receives all the return oil from the user 1.

User 2 is thus compulsory controlled – the speeds are according to the volume, the working pressures add together.

Circuit Diagrams

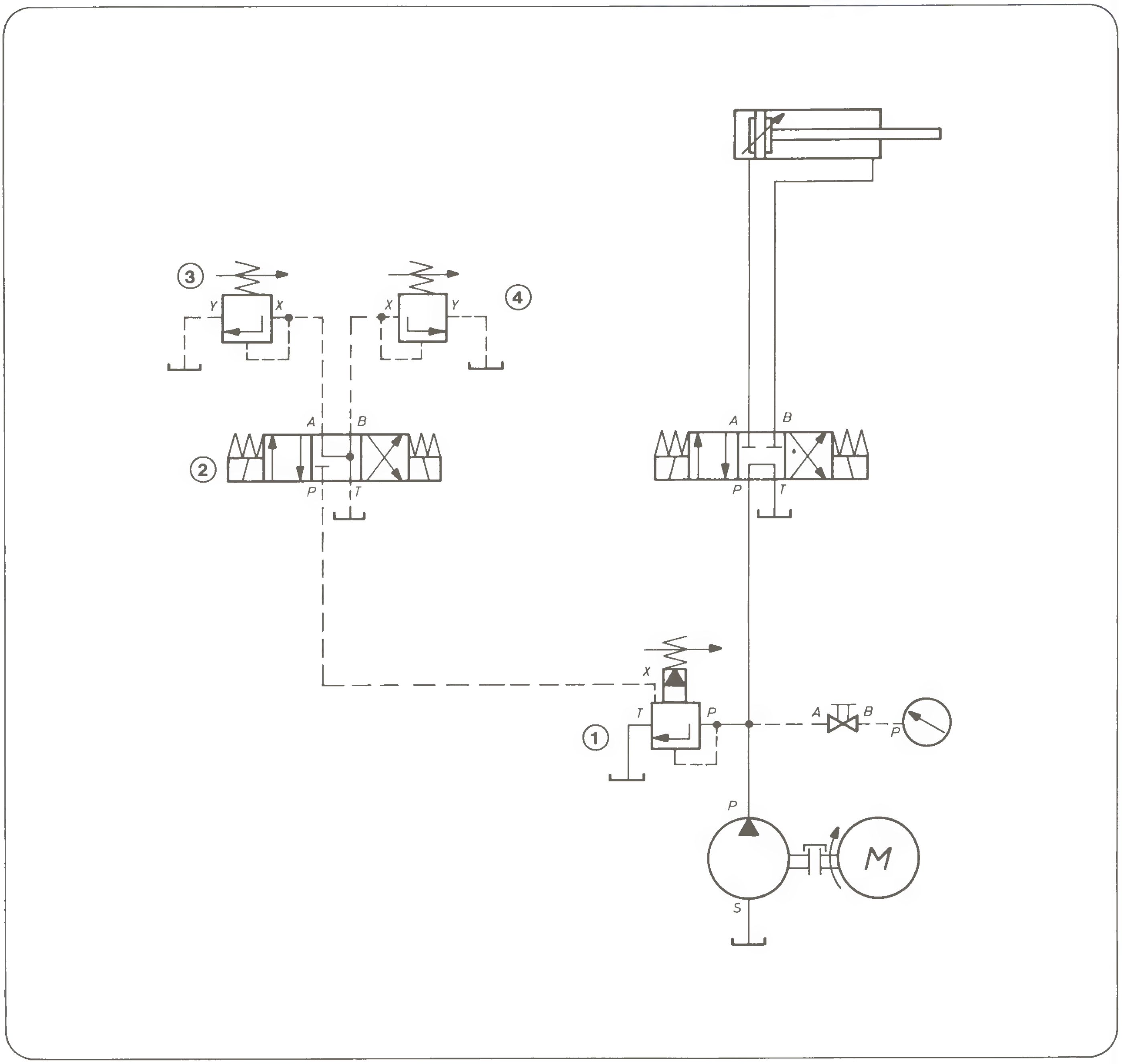
Hydraulic System with 3 Stage, Remote Controlled Pressure Relief

If, for example, a 3 way pressure stage is required on a hydraulic unit, this can be achieved by optionally switching 2 additional pressure relief valves or pilot valves.

The pilot operated relief valve 1 shown here can be connected to one or both pilot valves 3 or 4 by means of the directional control valve 2.

If directional control valve 2 is in centre position, valves 3 and 4 are connected to tank. Pressure in the system is that set at pressure relief valve 1.

If pressure control valve 3 or 4 (in this case, pilot valves) is actuated via the directional control valve 2, pressure at both valves 1 and 3 or 4 is effective. This means that the highest operating pressure must always be set at valve 1, so that the lower pressure set at the actuated pressure control valve 3 or 4, is set in the system. This arrangement can also be used as remote control.



Circuit Diagrams

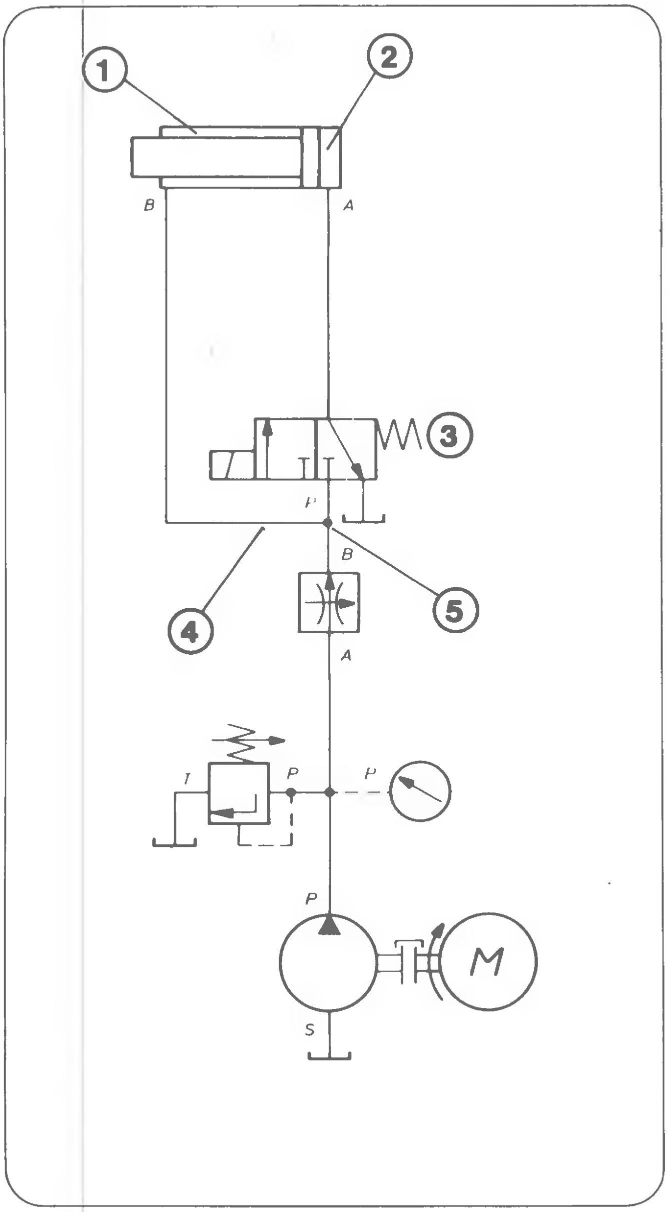
Hydraulic System with Differential Switching Cylinder

The so-called "differential switching" is used very frequently. A characteristic of this circuit is that the piston chamber 1 of the cylinder is continuously fed with oil, while the piston area 2 is affected by pressure or unloaded to tank, as required, by means of a 3 directional control valve 3. The force conditions affecting the piston rod correspond to the area ratios of the piston and rod sides, thus the name "differential switching".

This arrangement is used when hydraulic clamping is required and the pump should be as small as possible. When the piston rod travels out, the fluid 4 is taken from the rod chamber and fed, together with the pump flow 5 to the opposite side, i.e. into the spool chamber.

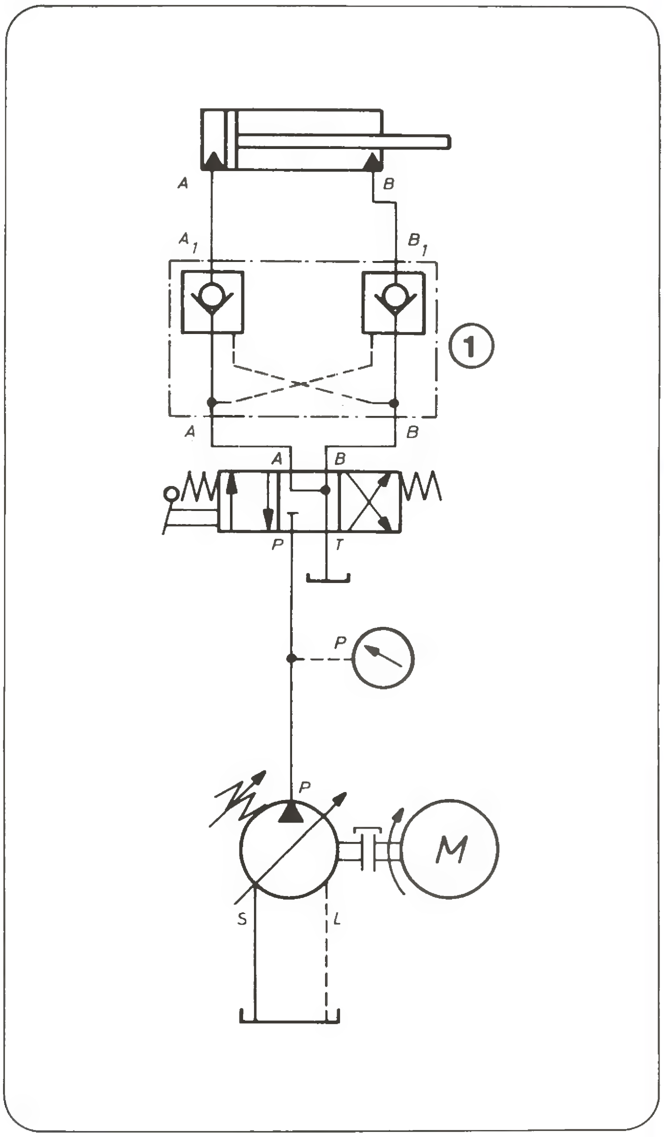
However, it must be noted that the force at the piston rod also corresponds only to the difference between the piston area and the piston annulus area, i.e. the piston rod area.

If we choose a surface area ratio of 1: 2 between the annulus area and the piston area, then we have in addition the advantage of equal forward and reverse speed of the differential cylinder piston rod.



Circuit Diagrams

Hydraulic System with Double Shut-off of one Cylinder



If a hydraulic cylinder has to be operated in both directions in such a way that it can be held in any position, a double opening check valve is used. In the directional control valve position shown, the cylinder cannot be moved by external forces in either direction.

Depending on the direction of the force, the left or right check valve blocks the flow hermetically.

The check valve arranged in the sequence is actuated from the supply side, to enable outward or inward travel of the cylinder.

Please note that, when the directional control valve is in centre position, both ports of the check valves must be unloaded to tank. Only then can a swift and perfect closure of the valve poppets and a hermetic seal be guaranteed.

Circuit Diagrams

Hydraulic System with Backpressure Valve and Check Valve

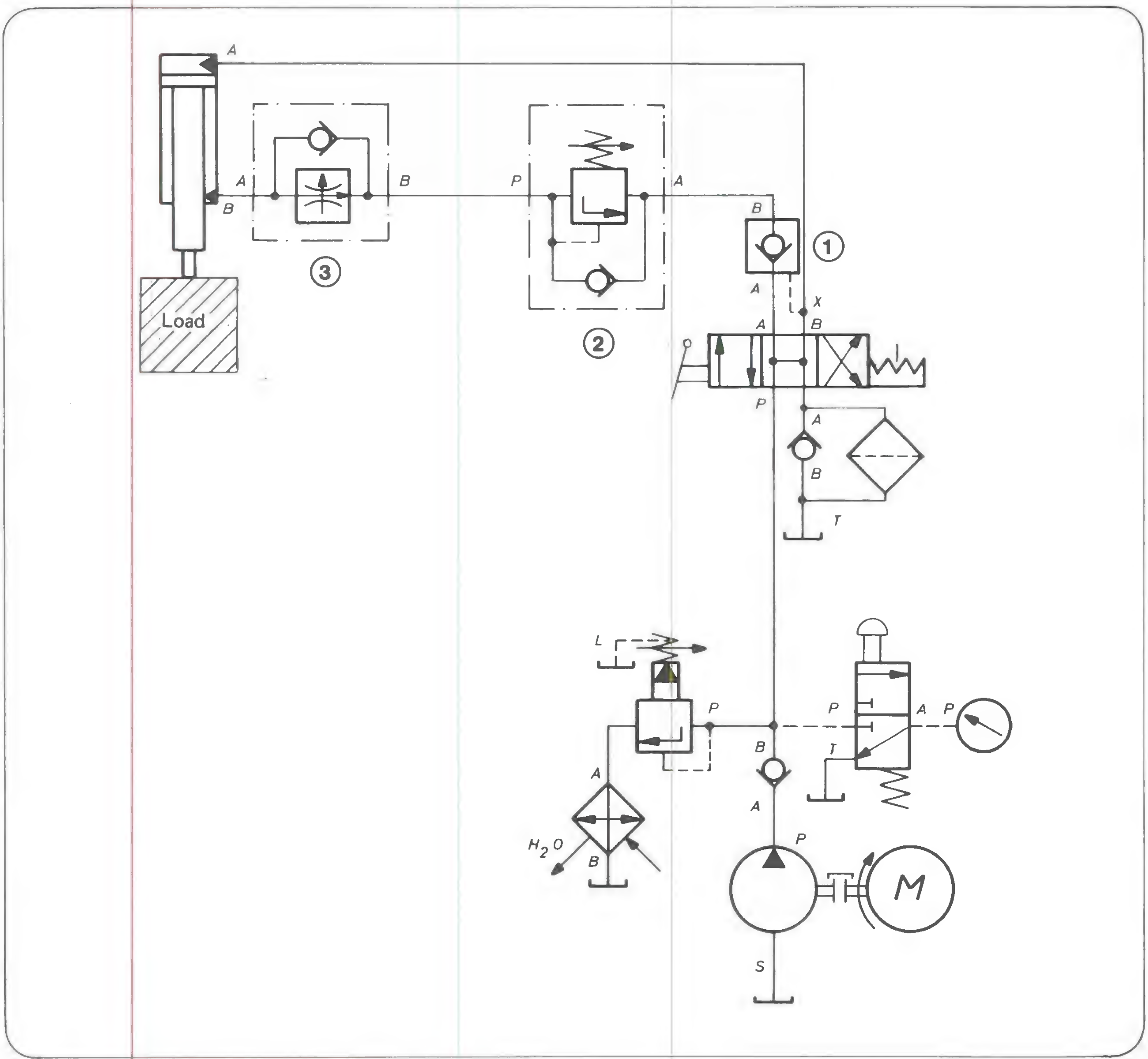
If a load affects a working cylinder continuously (e.g. the slide of a broaching machine), the cylinder must be protected from lowering due to leakage in the directional control valve. The pilot operated check valve 1 is built into the return line to act as such a protection.

A backpressure valve (pressure sequence valve 2) should also be provided. The cracking pressure of this valve is set at approximately 10% above the weight of the load to be held. We have here the effect of hydraulic backpressure.

This "initial stress value" is overcome and the downward movement induced only after pressurising the piston side (A).

The lowering speed is set using the flow control valve 3.

The check valve, arranged in parallel to it, allows fast lifting of the load.



Circuit Diagrams

Hydraulic System with Pressure Control at one Cylinder and Pressure-dependent Opposite Switching of a second Cylinder in the Forward or Return Line

We have here a very simple example of a chucking device with boring feed. In order to be quite sure that the workpiece is firmly clamped before the bore feed is actuated, a so-called "pressure related following or sequence circuit" is used.

The pedal operated 4/2 directional valve 1 is held in centre position by a spring. Both cylinders (boring and chucking) are in the inside position. If directional control valve 1 is moved to switching position, port P is connected with B and A with T.

Oil can flow directly into the chucking cylinder via the pressure control valve 2, opened in centre position. The cylinder then travels out. The connection to the boring cylinder is blocked by means of pressure sequence valve 3.

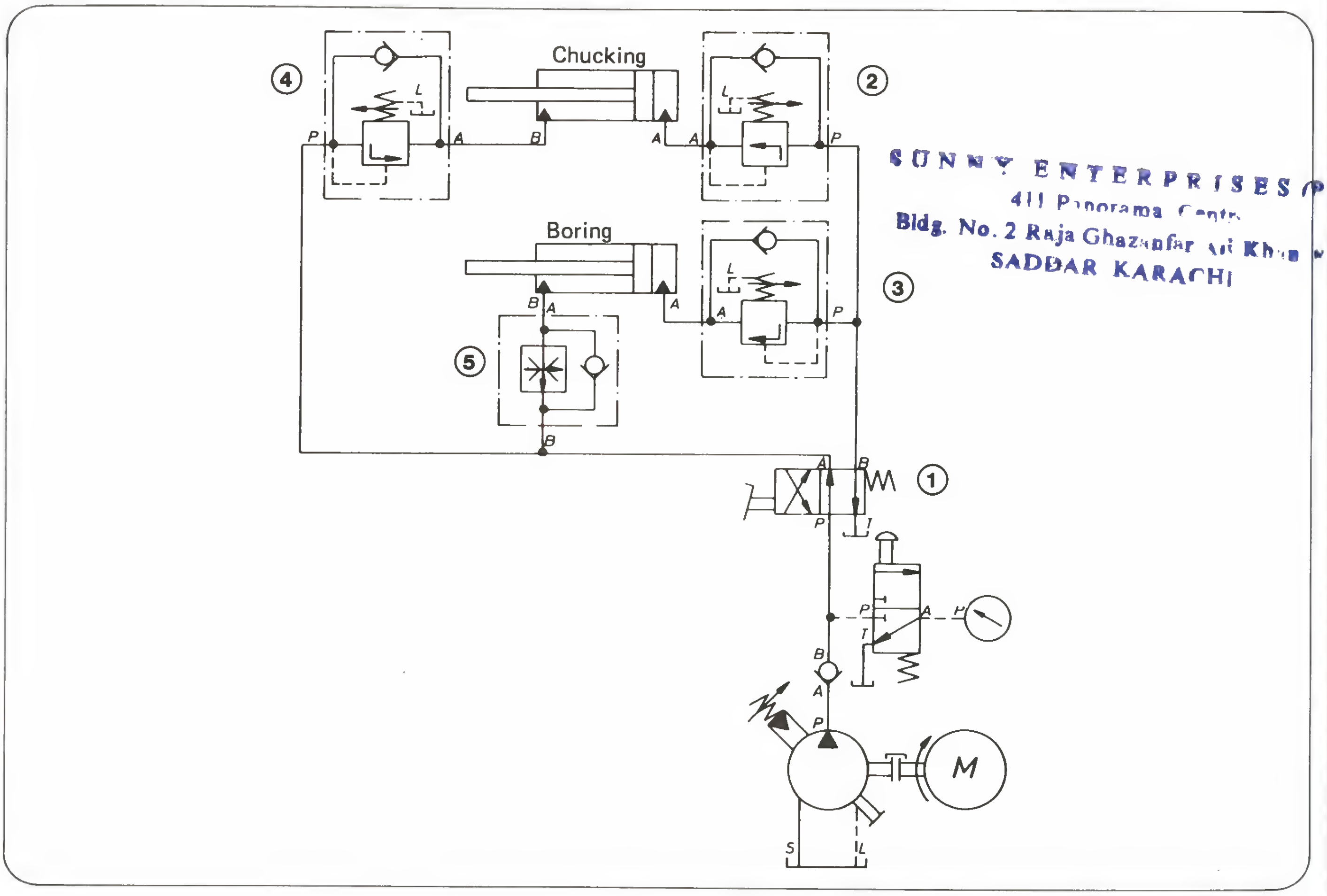
When the chucking cylinder has reached its position, pressure increases. Pressure at the chucking cylinder is that set at the pressure control valve 2. Pressure in the system from the pump to the pressure control valve continues to increase to the value set at the pressure sequence valve 3. When sequence pressure is reached, valve 3 opens. The boring cylinder travels out at the speed given by means of the flow control valve 5.

For the return of the cylinder, the sequence must be reversed. The chucking cylinder cannot release the workpiece until the boring cylinder is back in neutral position.

This sequence is determined by the pressure sequence valve 4. If the spring brings the directional control valve 1 into neutral position, return stroke is induced. Fluid reaches the boring cylinder and the connection to the chucking cylinder is blocked by means of pressure sequence valve 4.

If the boring cylinder is in end position, pressure increases. If the value set at valve 4 is reached, the valve opens the connection to the chucking cylinder, causing this to return.

In this case, a self-priming, pressure compensated pump with variable stroke volume is provided (e.g. vane pump type V3 — see chapter on pumps). The maximum operating pressure is therefore fixed at the pump.



Circuit Diagrams

Hydraulic System of a Press with Prefill Valve and Fast Forward Cylinder

High forces and therefore large volume cylinders are generally required for presses. So-called "prefill valves" are used, in order to avoid using expensive large volume pumps for the fast movements of the press piston. In principle, these are large size pilot operated check valves.

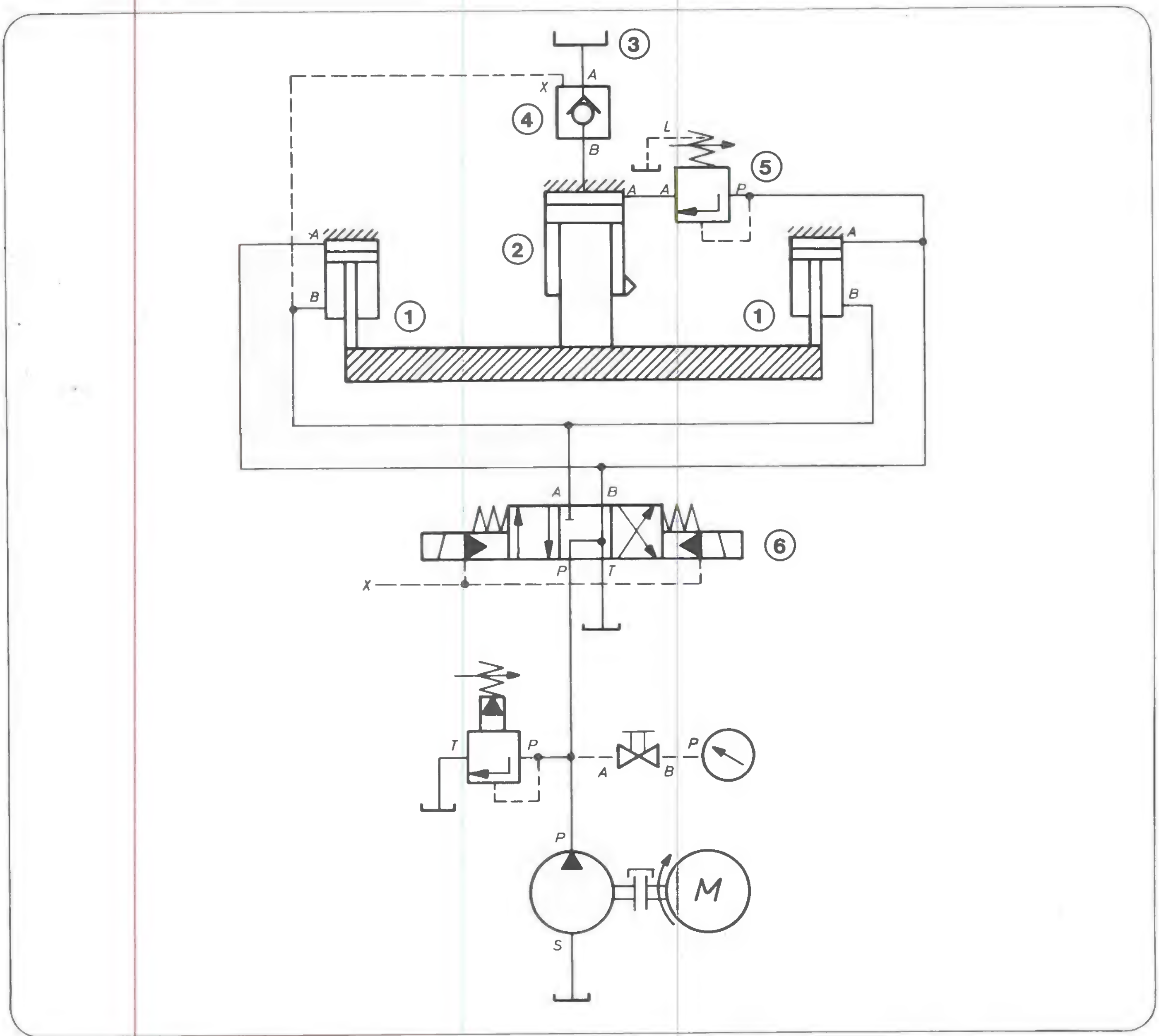
The method of operation is as follows:

Let us assume that the press ram is in its upper end position and that the downward movement is actuated by means of the 4/3 way directional valve 6 (crossed arrow position) when pressure affects both fast forward cylinders 1. The fluid required for the large press plunger follows from a high tank 3 via the pilot operated check valve 4.

After the press beam touches the workpiece, the resistance increases and pressure in the system rises. The adjustable pressure sequence valve 5 opens and fluid reaches the large cylinder chamber of the press ram.

All three piston areas are now affected by full pressure. The prefill valve is held in closed position towards the high tank. At return stroke, the piston chambers (A) of the fast forward cylinder are unloaded and the chambers (B) filled with fluid.

At the same time, the pressure at port X of the prefill valve is effective by means of the control line. The main poppet is opened and the oil pushed back to tank by means of a control cylinder.



Circuit Diagrams

Hydraulic System for the Hydraulic Synchronisation of several cylinders using the so-called "Bowden Cable".

The so-called "Bowden cable" is a very useful, though somewhat expensive method of hydraulic synchronisation.

Two cylinders of the same size with through piston rods are switched in series. The following cylinder therefore carries out the same movement as the first cylinder, which is affected by pump flow. However, as the two series switched cylinder chambers cause only one movement of the head of liquid, they would experience a gradual stroke movement without additional filling, due to internal and perhaps also external leakage.

In order to act against this undesirable but unavoidable effect, this "Bowden cable part" is connected to the pump flow or tank for a short time by operating the 4/3 way control valve 2 arranged on the right after each complete stroke.

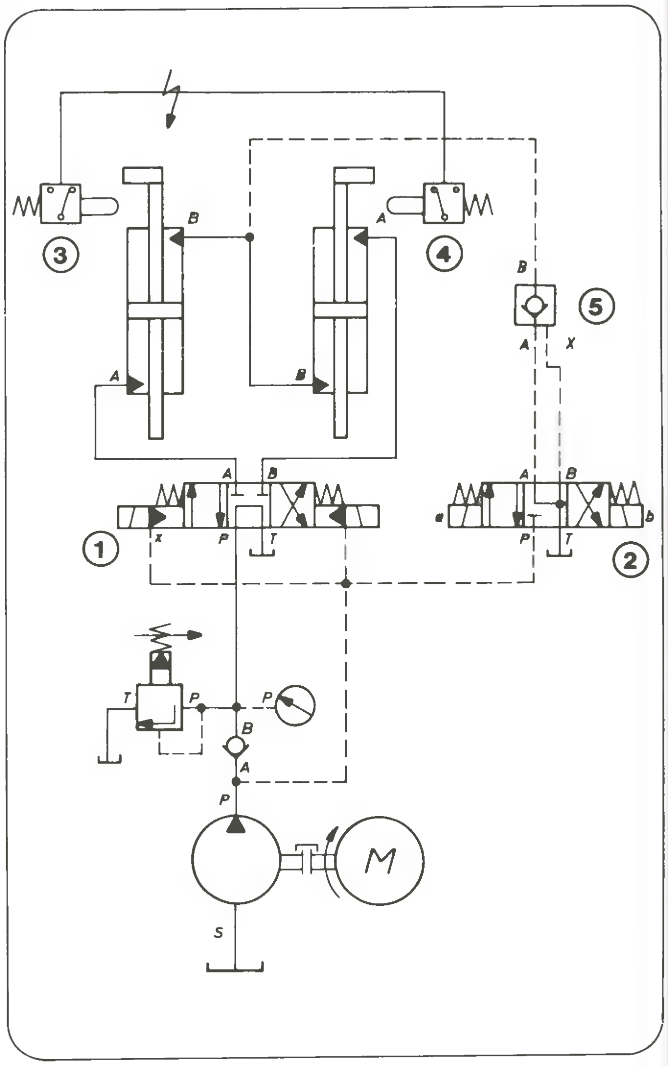
There are two reasons for unequal movement of the piston stroke:

a) The left-hand cylinder moves into the upper end position first, and operates the limit switch 3.
   Cause: too little fluid between the cylinders.
   Correction: Solenoid a is operated at directional control valve 2 by means of the left-hand limit switch 3. Pressure fluid is supplied via the control line until the right-hand cylinder also operates the limit switch. Solenoid a is thus de-energised.

b) The right-hand cylinder moves into the upper end position first, and operates the limit switch 4.
   Cause: too much fluid between the cylinders.
   Correction: Solenoid b at the directional control valve 2 is operated by means of the right-hand limit switch. The pilot operated check valve 5 is thus opened, and fluid drains until the left-hand cylinder is also in end position. Solenoid b is de-energised by means of the left-hand limit switch 3.

In this case, the even-ness of the piston stroke depends, not only on the points already mentioned, but also on the manufacturing process. It is a well known fact that two parts cannot be manufactured absolutely identical.

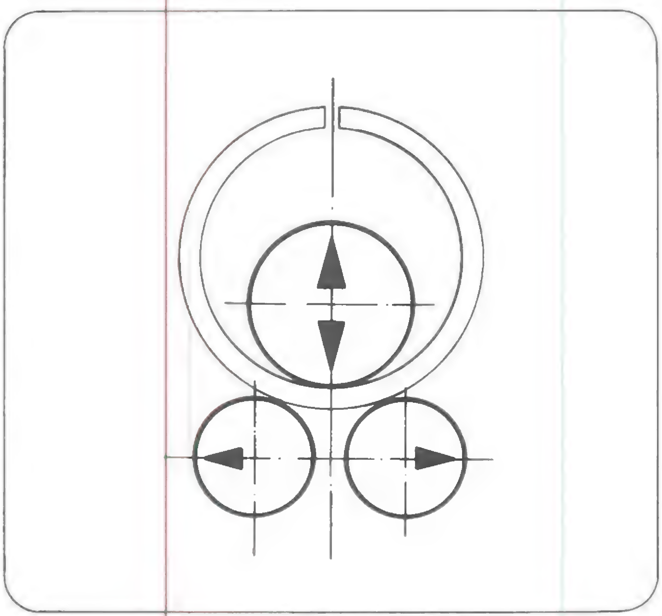
The directional control valve for filling up is also generally designed as a spool valve, and therefore certain leakage losses must be expected. It is therefore necessary to fit a pilot operated check valve 5 with a poppet.



Circuit Diagrams

Synchronising Control according to the Meter-Out Principle

The circuit diagram shows the synchronisation control for a roller on a 3 roll bending machine.



As an example, the two lower rollers in the diagram can be adjusted horizontally, and the upper roller vertically.

With this system, synchronisation is achieved by metering fluid out of the forward moving cylinder. Cylinders 12 and 13 must be supplied with oil separately by means of pumps 1 and 2, so that a certain metering-in is available. The two cylinders cannot influence each other.

Metering-out is via control valve 14.

In normal operation, the control valve must make up the difference which occurs due to:

— different pump flows

— oil compressibility

— different leakage

— play in the machine bearings

The direction in which the cylinders move is determined by means of directional control valves 8 and 9.

Pressure control valves 10 and 11 are used as back-pressure valves for outward travel.

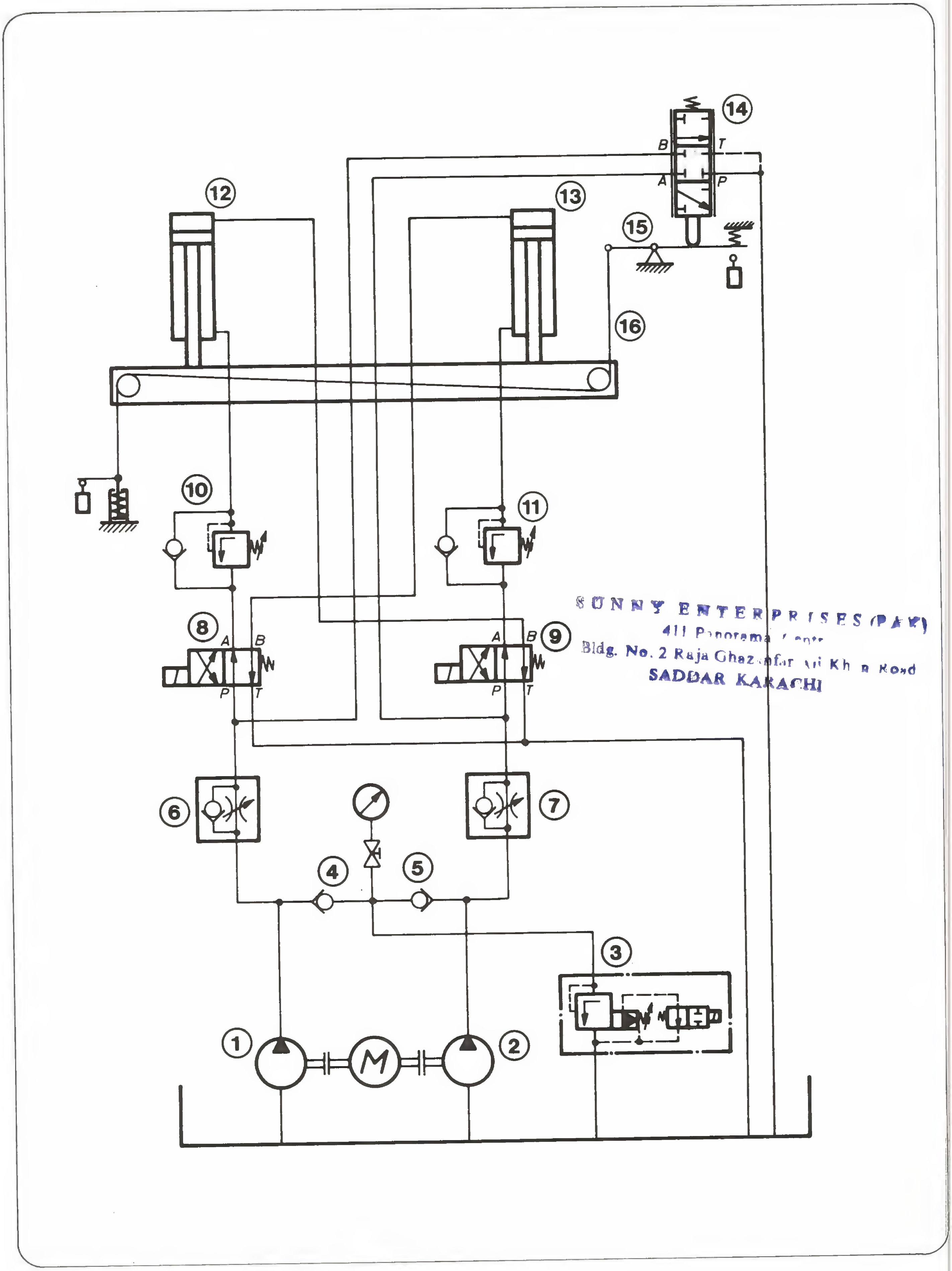
The quality of the synchronising valve depends basically on the error diagnosis. As shown in the diagram, it is preferable to control the regulating valve by means of a rocker 15. The rocker is operated by means of the mechanical synchronisation pick-up 16 (chain or steel band) and is arranged at the roller, as shown.

If one or other cylinder moves further forward, the position of the rocker alters accordingly by means of the operating element. The synchronising valve 14 is operated, the cylinder moving forwards is connected to the tank, and the oil thus metered out of the cylinder to the tank.

If, for example, the right-hand cylinder moves further forward at outward travel, the pick-up 16 pulls the rocker down and pushes the control valve 14 into switching position with connection B → P.

Synchronising control operates for both directions. This is achieved by connecting the synchronising valve with the supply line, e.g. in front of directional control valves 8 and 9, also by connecting ports A with the piston rod side of the relevant cylinder and ports B with the piston side of the other cylinder. The throttle check valves 6 and 7 serve to decompress the fluid at switch-over from pressing to return stroke.

Circuit Diagrams



Circuit Diagrams

Example of a Fork Lift Control

The circuit shown is designed as a control block.

The block contains 3 directional valve control spools in parallel switching series.

Directional valve 1 for stroke cylinder

Directional valve 2 for tilting cylinder

Directional valve 3 for additional hydraulics.

When the control elements (6 way valves) are in outlet position, there is bypass flow without pressure from pump connection P to tank connection T.

The users mentioned above are controlled by operation of the individual control elements (1, 2 and 3).

A type of flow divider is fitted additionally in the P line. It meters a preferred flow independent of load to the valves 2 (lowering) and 3 (additional hydraulics) at the set value.

This guarantees, for example, an exact lowering speed, even at high strokes. Flow reduction is low loss by means of flow control using the flow divider, since the pump pressure is only minimally higher than the user pressure.

The remaining flow is available for valve 1 (stroke cylinder) via a separate line (parallel switching). It flows to the tank when valve 1 is not actuated.

If valve 1 is actuated simultaneously with valve 2 or 3, the flow divider prevents the oil flows from influencing one another when user pressures differ.

If only valve 1 is actuated, the complete pump flow travels to the stroke cylinder via the bypass line. The flow divider is then outside the oil circuit.

Brake Valve for Forward Tilting (tilting valve)

A built-in brake valve can be fitted to port A2 or B2 of the control valve 2. It prevents undesirable acceleration of the load during forward tilting.

The brake valve spool allows the oil under load pressure to flow to the tank, as soon as pressure corresponding to the brake valve spring (approx. 30 bar) is available in the supply line.

Control pressure on starting decreases if more oil flows to the tank than corresponds to the supply quantity. The valve spring pushes the brake valve in closing direction, until there is a state of equilibrium.

This control circuit compensates the influence of the load size on the lowering speed.

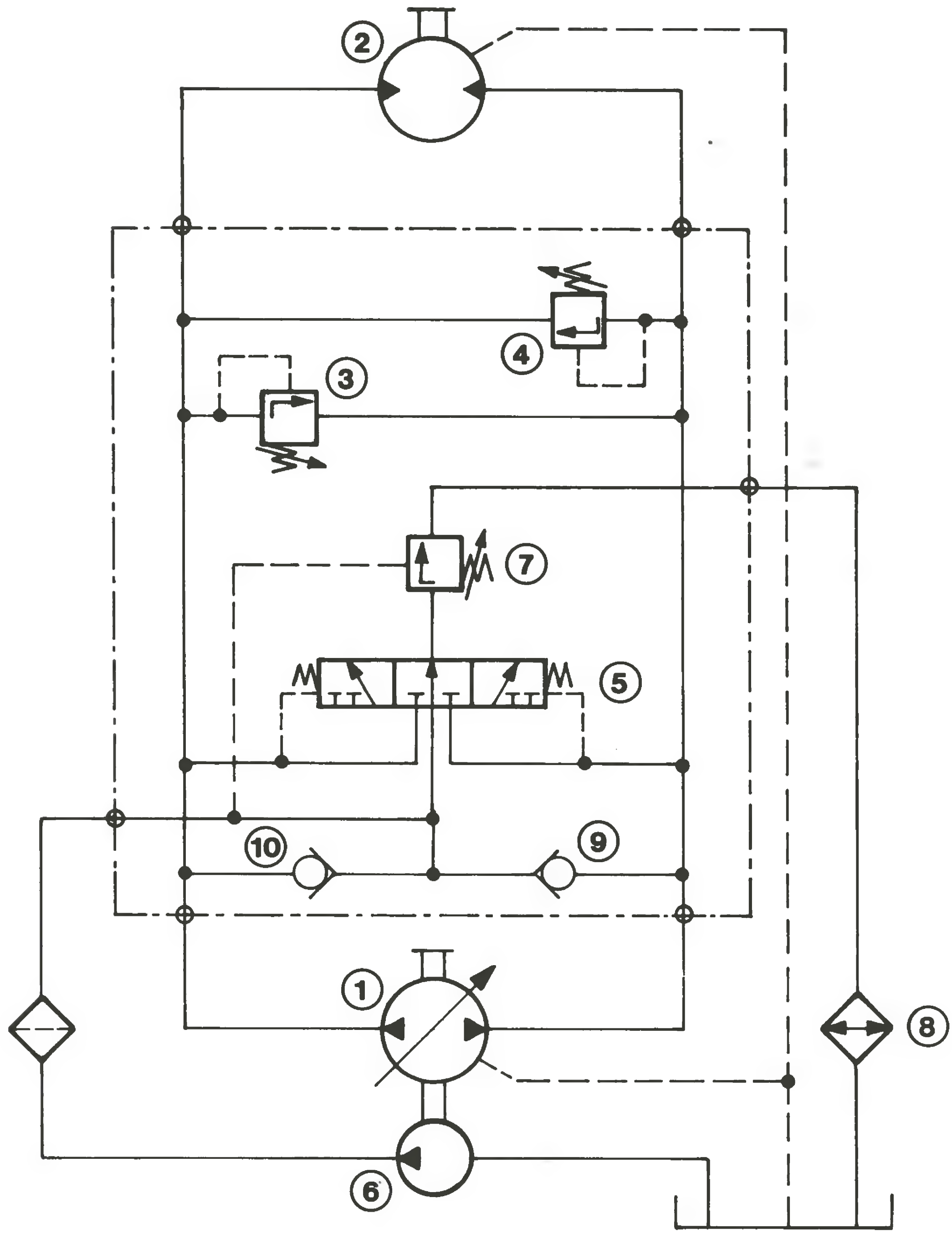
The brake valve therefore regulates the lowering speed independent of load, corresponding to the oil supplied.

Secondary pressure relief valves are fitted to the user ports of valves 1 and 3 in the example shown.

In order to operate the control elements, the spool ends on connection side A or B are fitted with a clevis or tongue piece, so that switching rods can be fitted. If levers are to be used, these pieces can be equipped with switching knobs.

A lever system can be fitted to the control spool to operate an electro switch, e.g. to switch on the el. motor on electro trucks. When the valves are operated, the oil motor and thus the pump are switched on.

Circuit Diagrams



Circuit Diagrams

Hydraulic System for Applications with Load Change

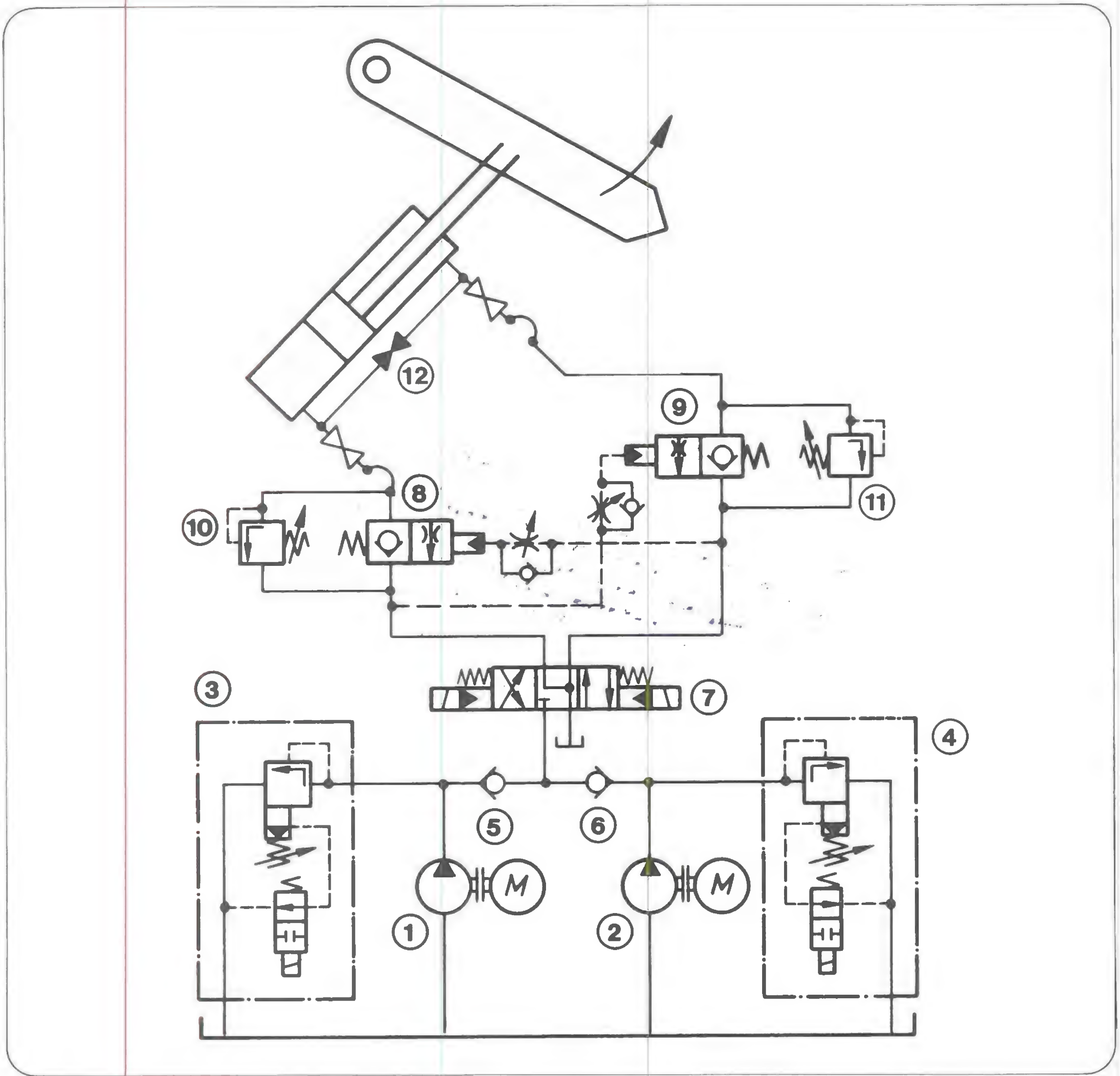
Load changes at the user can occur on drives for tilting devices, cluice gates, drawbridges, barriers, etc. This means that the effective direction of the load reverses during the movement sequence. Brake valves (8 and 9) are used in the example shown, so that the load does not cause the user to move more quickly than the speed given by the fluid quantity supplied.

The two fixed displacement pumps 1 and 2 supply oil into the pump line to the diectional control valve 7 via check valve 5 and 6.

The pumps can be switched to bypass without pressure by means of the built-on directional control valves for pressure unloading, using the pilot operated pressure relief valves 3 and 4.

If, for example, the cylinder should travel out, oil can be supplied freely via the brake valve 8. Brake valve 9 is initially controlled from the supply side. The initial control pressure decreases if the cylinder is moving too fast due to load change. The brake valve therefore moves in closing direction. The speed can thus be held constant independent of load.

Pressure relief valves 10 and 11 serve for secondary protection.



Formulae for Calculation

Hydraulic pumps

Flow $\boxed{Q = \frac{V \cdot n \cdot \eta_{vol}}{1000}}$ $[\frac{l}{min}]$

Drive power $\boxed{P_{an} = \frac{p \cdot Q}{600 \cdot \eta_{tot.}}}$ $[kW]$

Total output $\boxed{\eta_{tot.} = \eta_{vol} \cdot \eta_{hm}}$

Hydraulic motors

Flow $\boxed{Q = \frac{V \cdot n}{1000 \cdot \eta_{vol}}}$ $[\frac{l}{min}]$

Speed $\boxed{n = \frac{Q \cdot \eta_{vol} \cdot 1000}{V}}$ $[min^{-1}]$

Drive torque $\boxed{M_{ab} = \frac{\Delta p \cdot V \cdot \eta_{hm}}{2 \cdot \pi \cdot 100}}$ $[daNm]$

or $\boxed{M_{ab} = 1{,}59 \cdot V \cdot \Delta p \cdot \eta_{hm} \cdot 10^{-3}}$

$[daNm]$

Drive power $\boxed{P_{ab} = \frac{\Delta p \cdot Q \cdot \eta_{tot.}}{600}}$ $[kW]$

Hydraulic cylinders

geometric dimensions

Piston area $\boxed{A = \frac{d_1^2 \cdot \pi}{4 \cdot 100}}$ $[cm^2]$

or $\boxed{A = \frac{d_1^2 \cdot 0{,}785}{100}}$ $[cm^2]$

Piston rod area $\boxed{A_{St} = \frac{d_2^2 \cdot 0{,}785}{100}}$ $[cm^2]$

Q = Flow $[\frac{l}{min}]$

V = geometric flow (pump or motor) [cc]

n = drive speed of hydraulic pump $[\frac{rev}{min}]$ or min^{-1}

P_{an} = required drive power of the pump [kW]

p = operating pressure [bar or $\frac{daN}{cm^2}$]

$\eta_{tot.}$ = total output (~ 0.8 – 0.85)

η_{vol} = volumetric output (0.9 – 0.95)

η_{hm} = hydraulic-mechanical output (0.9 – 0.95)

M_{ab} = drive torque [daNm]

Δp = pressure drop between motor inlet and outlet [bar or $\frac{daN}{cm^2}$]

P_{ab} = drive power of hydraulic motor [kW]

d_1 = piston diameter (≙ cylinder dia) [mm]

d_2 = piston rod diameter [mm]

Formulae for Calculation

Piston annulus area =
piston area – piston rod area fläche

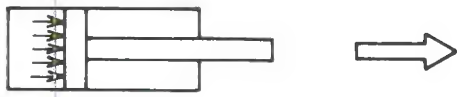
$$\boxed{A_R = \frac{(d_1^2 - d_2^2) \cdot 0{,}785}{100}}$$

[cm2]

Forces at the cylinder

Pressure force F_D

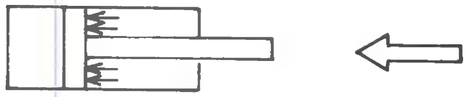
$$\boxed{F_D = \frac{p \cdot d_1^2 \cdot 0{,}785}{10\,000} \quad [kN]}$$



Traction force F_Z

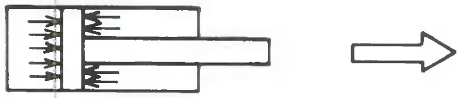
$$\boxed{F_Z = \frac{p \cdot (d_1^2 - d_2^2) \cdot 0{,}785}{10\,000}}$$

[kN]



Rod force F_S (≙ pressure force with differential switching)

$$\boxed{F_S = \frac{p \cdot d_2^2 \cdot 0{,}785}{10\,000} \quad [kN]}$$



The output of hydraulic cylinders
is approximately 0.85 – 0.95

Force $\boxed{F = p \cdot A}$ [daN]

Pressure $\boxed{p_{th} = \frac{F}{A}}$ [bar]

Stroke speed $\boxed{v = \frac{h}{t \cdot 1000}}$ $[\frac{m}{s}]$

$\boxed{v = \frac{Q}{A \cdot 6}}$ $[\frac{m}{s}]$

Required flow $\boxed{Q_{th} = A \cdot v \cdot 6}$ $[\frac{l}{min}]$

$\boxed{Q_{th} = \frac{V}{t} \cdot 60}$ $[\frac{l}{min}]$

$\boxed{Q = \frac{Q_{th}}{\eta_{vol.}}}$ $[\frac{l}{min}]$

F = force [daN]

p = operating pressure
[bar or $\frac{daN}{cm^2}$]

A = effective area [cm^2]

p_{th} = theoretical pressure without considering friction lcsses

v = stroke speed $[\frac{m}{s}]$

Q = flow $[\frac{l}{min}]$
taking leakage losses into consideration

Q_{th} = flow $[\frac{l}{min}]$ without leakage losses

η_{vol} = volumetric effeciency (~0.95)
taking leakage losses into consideration

Formulae for Calculation

Stroke volume $\boxed{V = \frac{A \cdot h}{10\,000}}$ [l]

Stroke time $\boxed{t = \frac{A \cdot h \cdot 6}{Q \cdot 1000}}$ [s]

V = stroke volume [l]

t = stroke time [sec]

h = stroke [mm]

Pressure losses in straight pipelines

$$\boxed{\Delta p = \lambda \cdot \frac{l \cdot \rho \cdot v^2 \cdot 10}{d \cdot 2}} \text{ [bar]}$$

Pipe friction coefficient for laminar flows

$$\boxed{\lambda_{lam.} = \frac{64}{R_e}}$$

Pipe friction coefficient for turbulent flows

$$\boxed{\lambda_{turb.} = \frac{0{,}316}{\sqrt[4]{R_e}}}$$

Reynolds number

$$\boxed{R_e = \frac{v \cdot d}{\nu} \cdot 10^3}$$

Flow speed

$$\boxed{v = \frac{Q}{6 \cdot d^2 \cdot \frac{\pi}{4}} \cdot 10^2}$$

Δp = pressure loss in straight pipeline (laminar or turbulent flow)

ρ = density $[\frac{kg}{dm^3}]$ ~0.89

λ = pipe friction coefficient

l = length of pipe [m]

v = flow speed $[\frac{m}{s}]$ in the line

d = internal diameter of pipe [mm]

ν = kinetic viscosity [cSt or $\frac{mm^2}{s}$]

Q = oil flow in the pipe $[\frac{l}{min}]$

Training and Test Units, Hydraulic Training

In order to accomodate the requirements of continuing technical developments, it is always necessary to carry out theoretical and practical instruction. The hydraulic industry is no exception to this. Skilled personnel must be available to carry out any service and maintenance work on equipment fitted with hydraulics. Training units are used to train engineers in these skills:

Hydraulic Training Unit (Rexroth)

Technical data:

size of valves to be fitted: size 6
operating pressure p_{max}: 50 bar
flow Q_{max}: 12 l/min
(variable displacement vane pump)

Fixed installations on this training unit are: motor/pump group with pressure gauge and a pressure relief valve set at 50 bar, return line with filter, leakage line, return line with measuring glass, thermometer, fine pressure gauge to be fitted separately, also a cylinder as a user.

Jsing cable and rollers, the cylinder arranged on the ide can be loaded in both directions of movement.

he individual pieces of hydraulic equipment are ted on changeable plates, which are hung on the front of the unit and are held by grooved pins. All components and line connections (P and T) are fitted with quick lock couplings. The line connections are made at the rear, using high pressure hoses.

In this way, the components used can be changed easily and quickly, and the line connections can be made as desired.

Hydraulic Training unit (Rexroth)

Technical data:

size of valves to be fitted: size 6, size 10, size 16
fixed displacement vane pump, type V2
p_{max} = 175 bar; Q_{max} = 10 l/min
variable displacement vane pump
p_{max} = 100 bar; Q_{max} = 12 l/min

The components are fitted and the line connections made in the same way as on the smaller unit. However, on this unit, directional valves can also be controlled electrically. A programmed sequence, controlled via limit switch and cylinder, is also fitted as standard. The hydraulic training unit is also designed for use as a test unit.

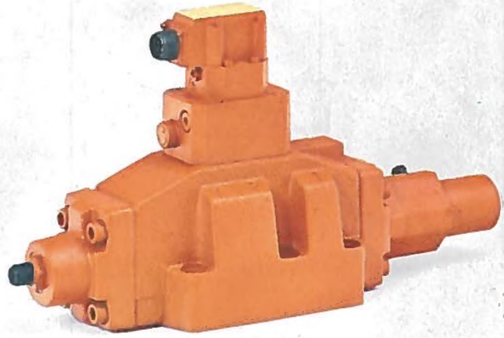
Summary of Rexroth Hydraulic Training

- specialist conferences for the individual branches
- practical training (HTP)
- general hydraulic course (HTQ)
- hydraulics in theory and practice (HTT)

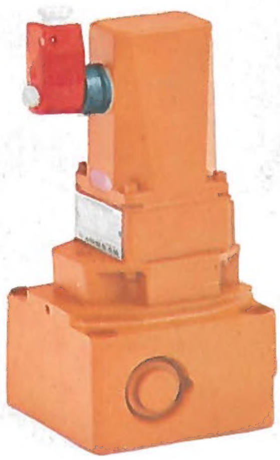
Training material:
set of colour slides and set of colour transparencies for overhead projector, showing sectional diagr...

Hydraulics for Pressure and Injection Moulding Machines

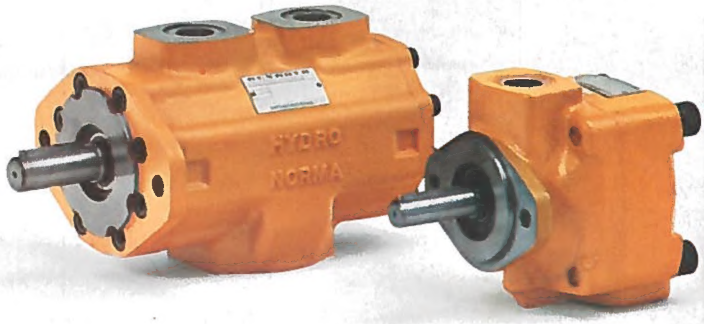
2/2 way cartridge valves



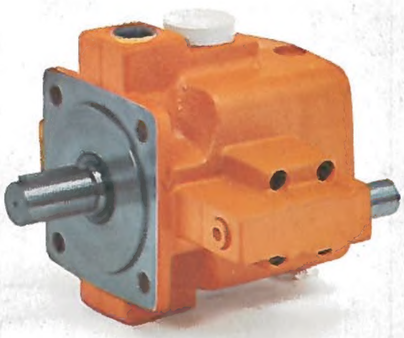
Directional proportional valve,
3 stage, el. feedback



Electrically operated
flow control valve
(size 10 and 16)



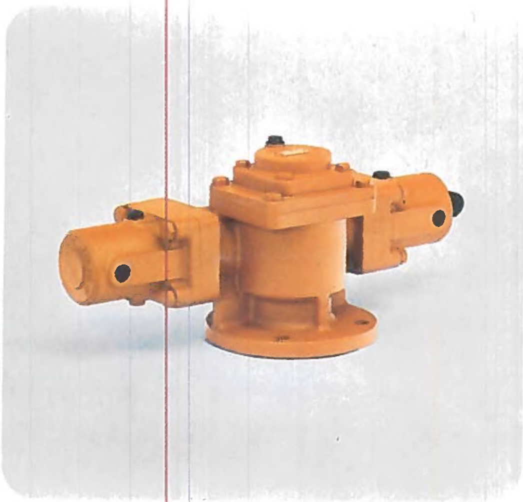
Fixed displacement vane pump type V2



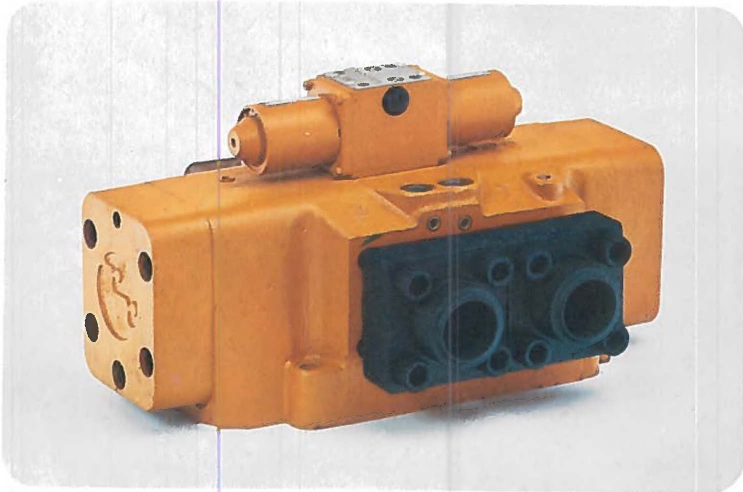
Variable displacement vane pump type V4

Pressure proportional valves for
pressure relief, pressure reduction
(size 10 to size 32)

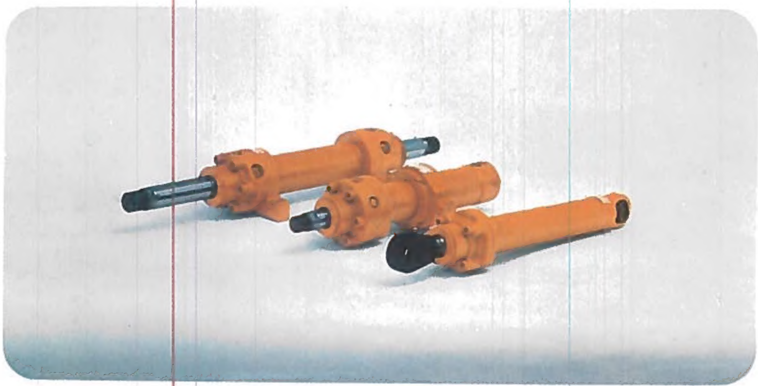
Hydraulics for Civil Engineering



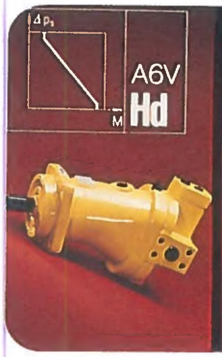
Actuator:
Md up to 2650 da Nm, p up to 200 bar



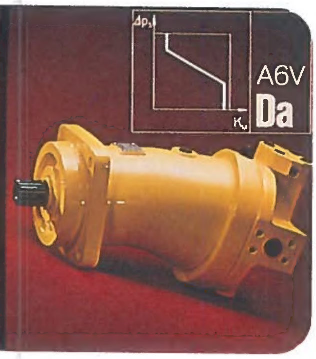
Directional spool valves for flange connections (up to size 102)
for subplate mounting (up to size 82)



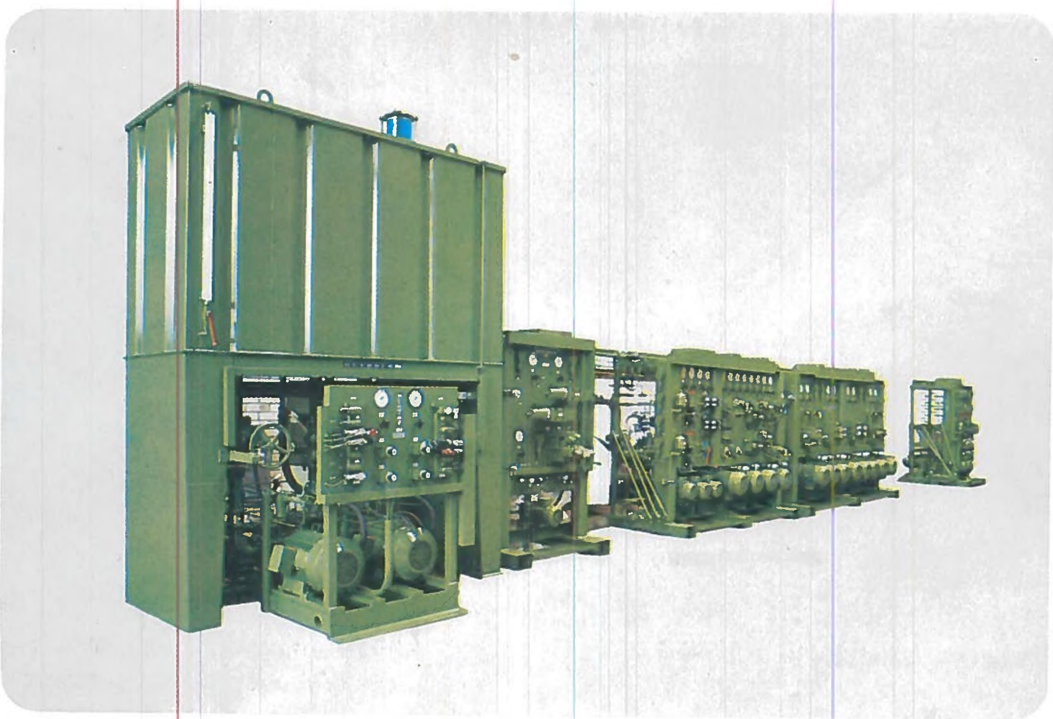
Hydraulic cylinder,
stroke up to 10 000 mm, pressure up to 400 bar

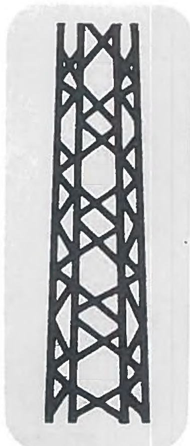


Variable displacement motor A6V with hydraulic control, pilot pressure related



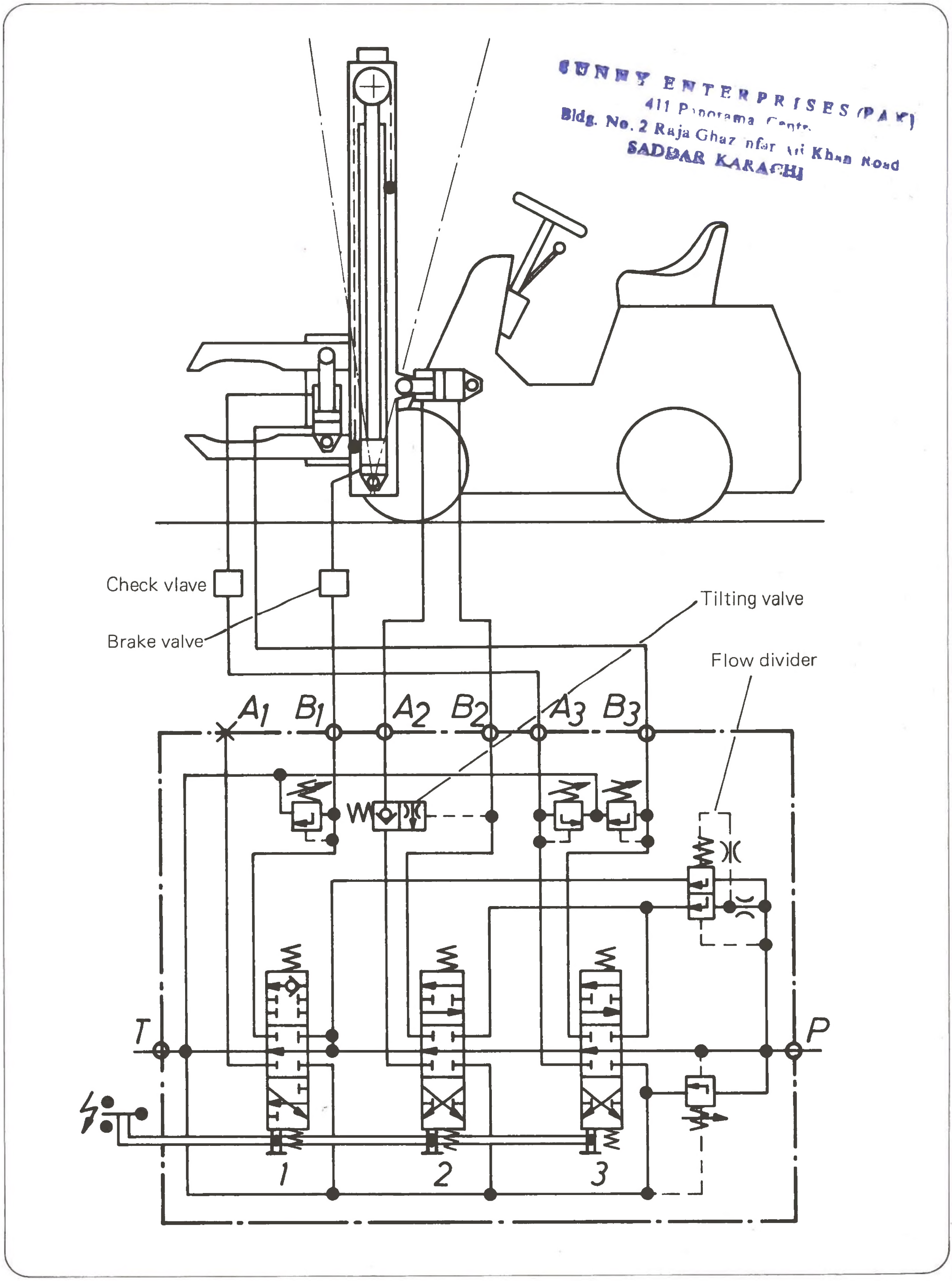
Variable displacement motor A6V with hydraulic control, speed related





Drive station for floodgates Stör

Circuit Diagrams



Circuit Diagrams

Closed Circuit

By "closed circuit" we mean a system which comprises a pump 1 and motor 2 and where the fluid circulates from the pump via the motor back into the pump suction line.

In most cases, the pump has variable displacement and is designed for both flow directions.

Only a few additional elements are necessary for practical operation of a closed circuit.

- Pressure Protection
  The two adjustable pressure relief valves 3 and 4 limit pressure in the high pressure side and protect the circuit from overloading. The drain fluid is fed to the low pressure side. The pressure control valves also serve to brake the motor at zero pump flow.

- Flushing Valve and Boost Circuit
  The flushing valve 5 is a hydraulically operated directional control valve. If pump 1 has zero flow, boost pump 6 delivers oil via the flushing valve 5 in centre position, the pressure relief valve 7 and the cooler 8 to the tank.

The boost or low pressure is set at pressure control valve 7. It is generally 8 ... 15 bar.

If the pump is delivering, i.e. oil is fed to the hydraulic motor, then the high pressure side (working pressure) operates the flushing valve. It opens the connection from the low pressure side to the pressure control valve 7.

Let us assume that the left-hand side is the high pressure side (motor 2 clockwise rotating); flushing valve 5 is then switched to the right via the left-hand control line. The low pressure side (on the right) is thus connected with the pressure control valve 7, which is balanced out by the boost pump.

Oil flows from the low pressure side via the flushing valve 5 and the pressure relief valve 7 back to tank. At the same time, the boost pump 6 delivers oil to the low pressure side via check valve 9. Check valve 10 is held closed from the high pressure side.

If the flow direction of the variable displacement pump is changed, the flushing valve is pressurised from the opposite side. The flushing process can take place correspondingly.

Heat dissipation and change of fluid in closed circuit is thus achieved, using a flushing valve.